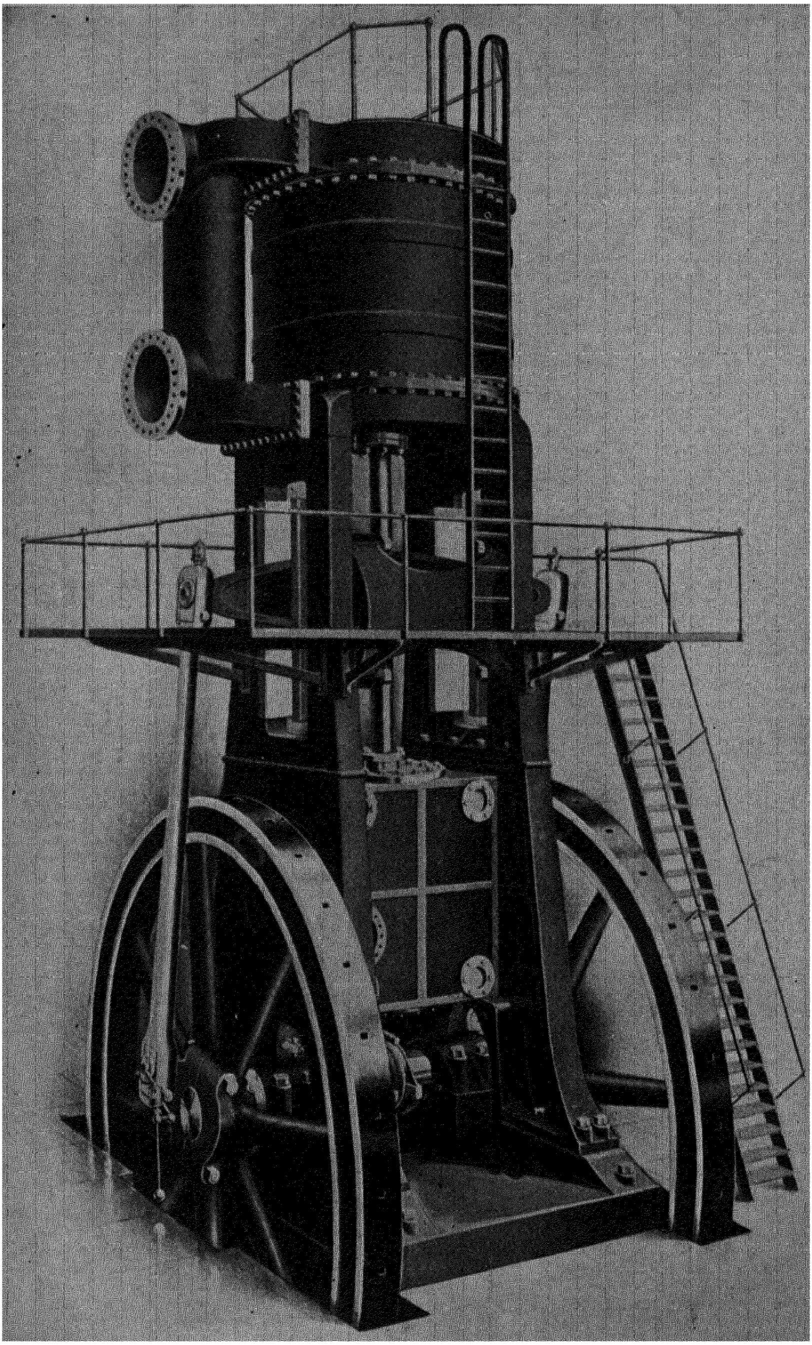

TOD VERTICAL LONG-CROSSHEAD BLOWING ENGINE, EQUIPPED WITH PATENTED PLATE AIR VALVES

Courtesy of The William Tod Company, Youngstown, Ohio

Cyclopedia

of

Engineering

VOL II

A General Reference Work on

STEAM BOILERS AND PUMPS, STEAM, STATIONARY, LOCOMOTIVE, AND MARINE ENGINES, STEAM TURBINES, GAS AND OIL ENGINES, GAS-PRODUCERS; COMPRESSED AIR, REFRIGERATION, ELEVATORS, HEATING AND VENTILATION, MANAGEMENT OF DYNAMO-ELECTRIC MACHINERY, POWER STATIONS; ETC.

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Grateful acknowledgment is made here also for the invaluable co-operation of the foremost engineering firms in making these volumes thoroughly representative of the best and latest practice in the design and construction of steam and electrical machines; also for the valuable drawings and data, suggestions, criticisms, and other courtesies.

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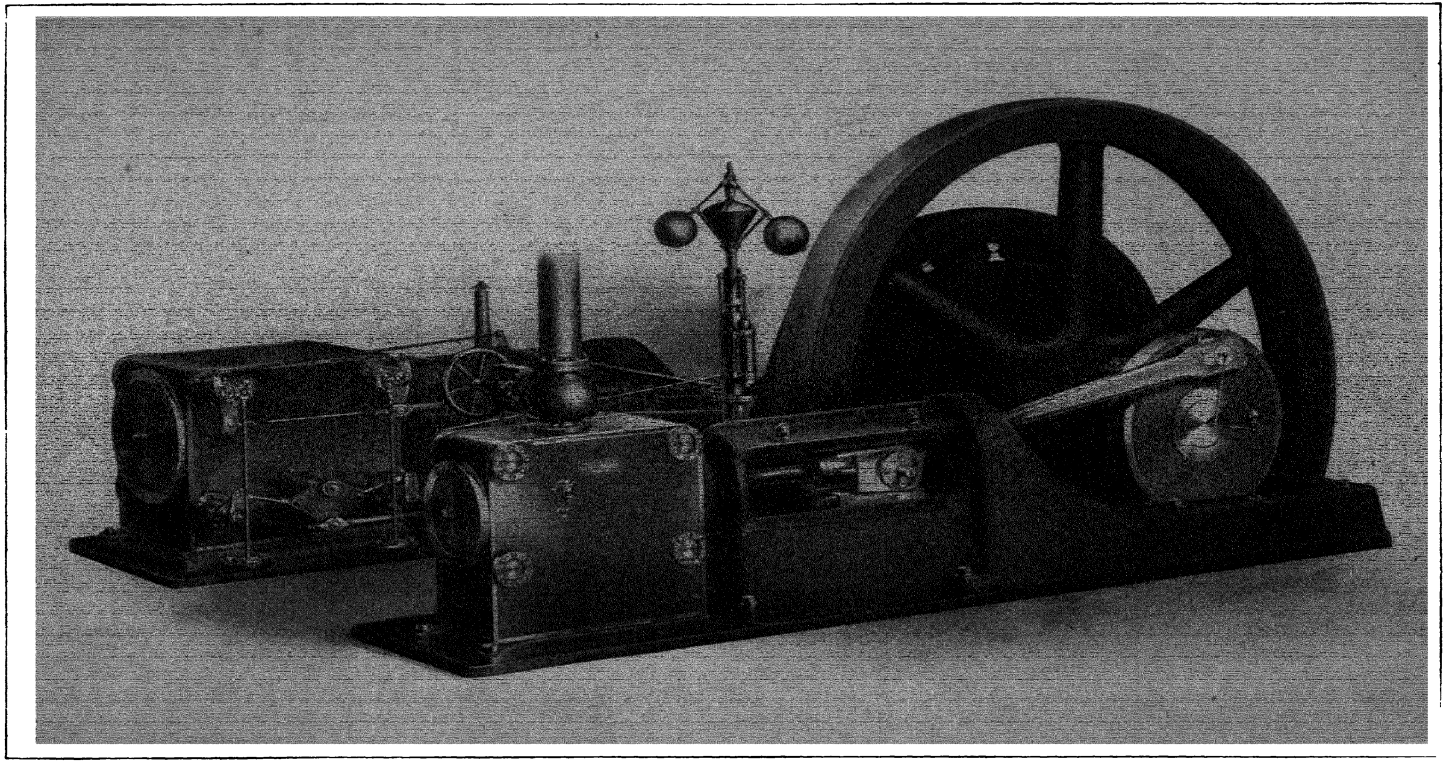

HEAVY DUTY, CROSS-COMPOUND, DIRECT-CONNECTED CORLISS ENGINE (FULL TANGYE BED)
Courtesy of C. & G. Cooper Company

Foreword

THE "prime mover", whether it be a massive, majestic Corliss, a rapidly rotating steam turbine, or an iron "greyhound" drawing the Limited, is a work of mechanical art which commands the admiration of everyone. And yet, the complicated mechanisms are so efficiently designed and everything works so noiselessly, that we lose sight of the wonderful theoretical and mechanical development which was necessary to bring these machines to their present state of perfection. Notwithstanding the genius of Watt, which was so great that his basic conception of the steam engine and many of his inventions in connection with it exist today practically as he gave them to the world over a hundred years ago, yet the mechanics of his time could not build engine cylinders nearer true than three-eighths of an inch—the error in the modern engine cylinders must not be greater than two-thousandths of an inch.

¶ But the developments did not stop with Watt. The little refinements brought about by the careful study of the theory of the heat engine; the reduction in heat losses; the use of superheated steam; the idea of compound expansion; the development of the Stephenson and Walschaert valve gears—all have contributed toward making the steam engine almost mechanically perfect and as efficient as is inherently possible.

¶ The development of the steam turbine within recent years has opened up a new field of engineering, and the adoption of this form of prime mover in so many stationary plants like the immense Fisk Station of the Commonwealth Edison Company, as well as its use on the gigantic ocean liners like the Lusitania, makes this angle of steam engineering of especial interest.

¶ Adding to this the wonderful advance in the gas engine field — not only in the automobile type where requirements of lightness, speed, and reliability under trying conditions have developed a most perfect mechanism, but in the stationary type which has so many fields of application in competition with its steam-driven brother as well as in fields where the latter can not be of service — you have a brief survey of the almost unprecedented development in this most fascinating branch of Engineering.

¶ This story has been developed in these volumes from the historical standpoint and along sound theoretical and practical lines. It is absorbingly interesting and instructive to the stationary engineer and also to all who wish to follow modern engineering development. The formulas of higher mathematics have been avoided as far as possible, and every care has been exercised to elucidate the text by abundant and appropriate illustrations.

¶ The Cyclopedia has been compiled with the idea of making it a work thoroughly technical, yet easily comprehendible by the man who has but little time in which to acquaint himself with the fundamental branches of practical engineering. If, therefore, it should benefit any of the large number of workers who need, yet lack, technical training, the publishers will feel that its mission has been accomplished.

¶ Grateful acknowledgment is due the corps of authors and collaborators — engineers and designers of wide practical experience, and teachers of well-recognized ability — without whose co-operation this work would have been impossible.

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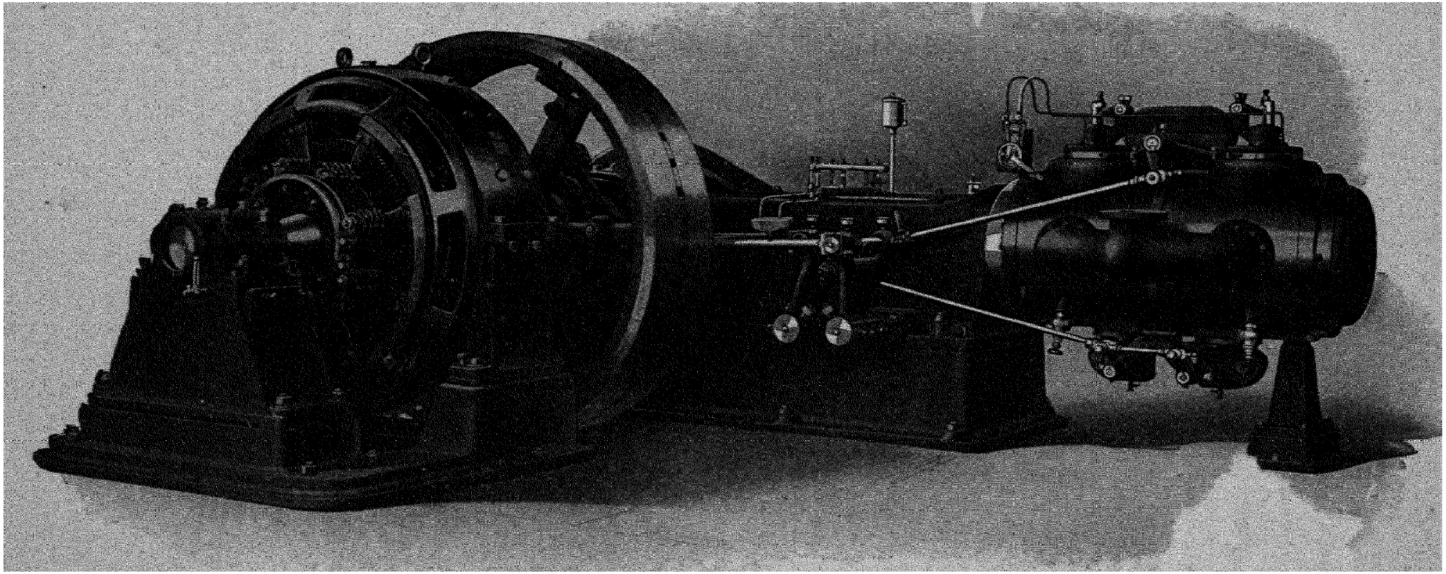
† For professional standing of authors, see list of Authors and Collaborators at front of volume.



SIDE CRANK UNIVERSAL-UNAFLOW ENGINE DIRECT-CONNECTED TO GENERA

Courtesy o' S .nner Engine Compan P insylvania

STEAM ENGINES

PART I

DEVELOPMENT

Early History. In the study of this subject, it is thought advisable to review the historical development of the steam engine in order that a broad conception of it may be obtained. It is not intended, however, to give the history of the steam engine in detail—although it is an exceedingly interesting one, which would be beneficial for any one to review—but rather, a short résumé in order that the student may be prepared for a detailed study of the modern engine.

The first steam engines of which we have any knowledge were described by Hero of Alexandria, in a book written two centuries before Christ. Some of them were very ingenious, but the best were little more than toys. From the time of Hero until the seventeenth century little progress was made. At this time, however, there was a great need of steam pumps to remove water from the coal mines. In 1615, Salomon de Caus devised an arrangement, consisting of a vessel having a pipe leading from the bottom which was filled with water and then closed. When heat was applied to the vessel, steam was formed, which forced the water through the discharge pipe.

Later an engine was constructed in the form of a steam turbine, but was unsuccessful, and the attention of the inventors was again turned to pumps.

Savery. Finally Thomas Savery completed, in 1693, the first commercially successful steam engine. It was very wasteful of steam as compared with engines of today but, as being the first engine to accomplish its task, it was successful. Savery's engine, Fig. 1, consisted of two oval vessels A_1 and A_2, placed side by side and in communication with a boiler B_1. The lower parts were con-

nected by tubes fitted with suitable valves. In operation, steam from the boiler was admitted, say, to the vessel A_2 and the air driven out. The steam was then condensed and a vacuum formed by letting water play over the surface of the vessel. When valve *1* was opened, this vacuum drew water from below until the vessel was full. The valve was then closed and steam again admitted by valve *2*, so that on opening valve *3* the water was forced out through the delivery pipe *C*. The two vessels worked alternately. When one was filling with water, the other was open to the boiler and was being emptied. Of the two boilers B_1 and B_2, one supplied steam to the oval vessels and the other was used for feeding water

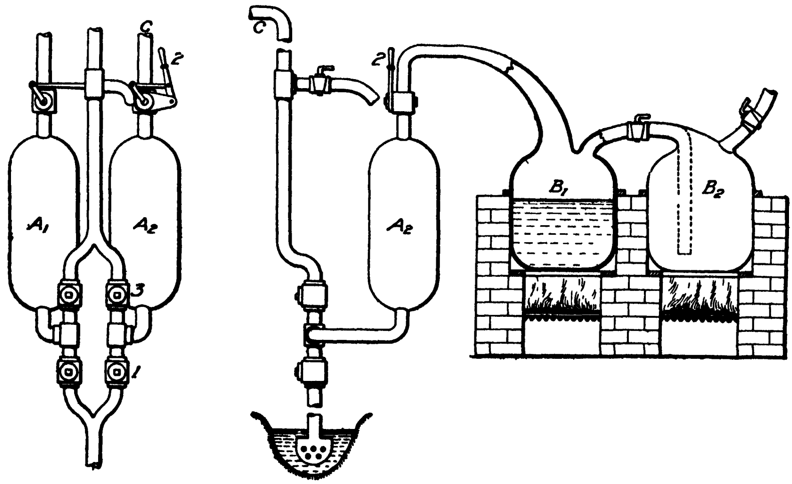


Fig 1. Early Form of Steam Pumping Engine

to the first boiler. In operation the second boiler was filled while cold, and after a fire had been lighted under it, acted like the vessel used by Salomon de Caus and forced a supply of feed water into the main boiler.

A modification of Savery's engine—the pulsometer shown in Fig. 2—is still found in use in places where an ordinary pump could not be used and where extreme simplicity is of especial advantage. Its valves work automatically and it requires very little attention.

A serious difficulty with Savery's engine resulted from the fact that the height to which water could be raised was limited by the pressure which the vessels could sustain. Where the mine was

very deep it was necessary to use several engines, each one raising the water a part of the whole distance. The consumption of coal in proportion to the work done was about twenty times as great as that of a good modern steam engine. This was largely, though not entirely, due to the immense amount of steam which was wasted by condensation when it came in contact with the water in the oval vessels.

Newcomen. The next great step in the development of the steam engine was taken by Newcomen, who in 1705 succeeded in developing a scheme which prevented contact between the steam and the water to be pumped, thus diminishing the amount of steam uselessly condensed. He introduced the first successful engine which used a piston working in a cylinder.

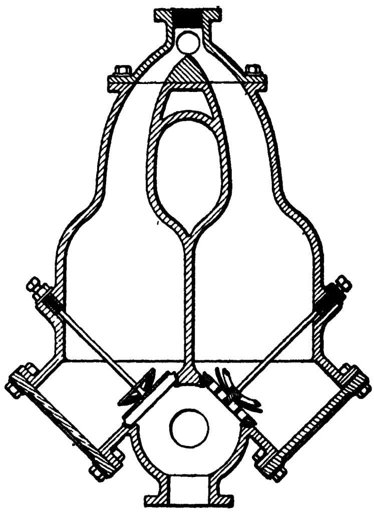


Fig 2 Pulsometer

In Newcomen's engine, Fig. 3, there was a horizontal lever *A*, pivoted at the center, carrying at one end a long heavy rod *B* which connected with a pump in the mine below. A piston *C* was hung from the other end of the lever and worked up and down in a vertical cylinder *D*, which was open at the top. Steam acting on the lower side of the piston, at atmospheric pressure, was admitted from the boiler to the cylinder, and as the pressure was the same both above and below the piston, the weight of the heavy pump rod raised the piston. A jet of water in the cylinder condensed the steam and formed a vacuum. This left the piston with atmospheric pressure above and very little pressure below (a partial vacuum), so it was forced down and the pump rod raised again. Steam could again be admitted to the cylinder; the pump rod would fall; and the process could be continued indefinitely.

In the days of Newcomen it was very difficult to obtain good workmanship. For this reason it was often necessary to make the cylinders of wood. In order to prevent steam from blowing

around the piston, or air from leaking in where steam was being condensed, it was customary to keep a jet of water playing on the top of the piston.

One great trouble with all of these engines was that some one was required to open and close the cocks, and boys were generally employed to do this work. One boy, in order to get time to play, rigged a catch at the end of a cord which was attached to the beam

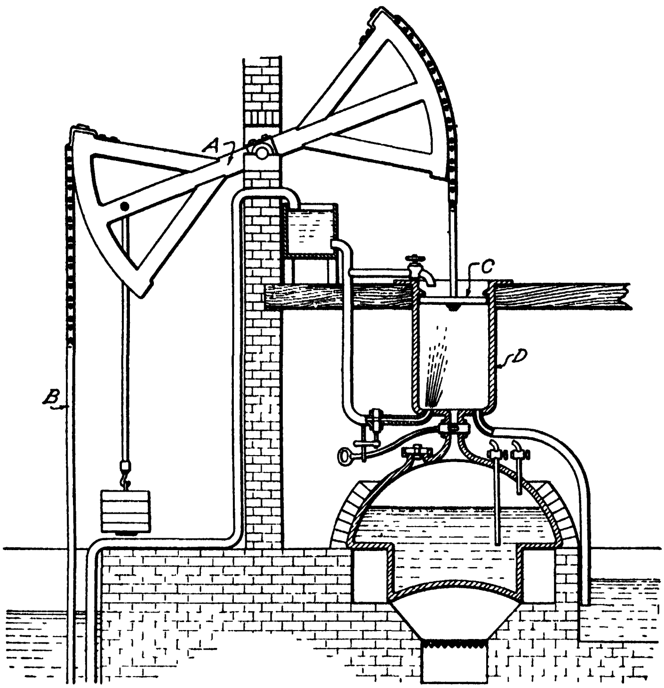


Fig 3 Newcomen's Steam Pumping Engine

overhead, and this did the work for him. By making the valves in this way automatic, made it possible to dispense with the services of the boy and at the same time greatly increase the speed of the engine.

The Newcomen engine was improved slightly from time to time by different inventors and was very extensively used until the time of Watt, a very few of them still being in existence today. While this engine was a success and a great improvement over its

predecessors, it was still very large, wasteful, and heavy in comparison with the work done, and the cylinders, when made of iron, were simply cast and not bored, thus leaving a rough, inner wall.

Watt. In the year 1763, a small model of a Newcomen engine was taken to the shop of an instrument maker in Glasgow, Scotland, to be repaired. This instrument maker, whose name was James Watt, had been studying steam engines for some time and he became very interested in this model. He was a man of great genius, and before he died his inventions had made the steam engine so perfect a machine that there has been but one really great improvement in it since his time, namely, compounding.

He found that to obtain the best results it was necessary, "*first*, that the temperature of the cylinder should always be the same as that of the steam which entered it; and *second*, that when steam was condensed it should be cooled to as low a temperature as possible." All improvements in steam-engine efficiency have been in the direction of a more complete realization of these two conditions.

In order to keep the cylinder nearly as hot as the entering steam, Watt no longer injected water into the cylinder to condense the steam, but used a separate vessel or condenser. He made his piston tight by using greater care in construction, so that it was unnecessary to have a water seal at the top. He then covered the top of the cylinder to prevent air from cooling the piston. When this was done he could use steam above the piston as well as below; this made the engine double acting.

Also, in the effort to keep the cylinder as hot as the entering steam, he enclosed the cylinder in a larger one and filled the space between with steam. This was not often done, however, and only of late years has the steam jacket been of much advantage. Also, the steam was used expansively, that is, the admission of steam was stopped when the piston had made a part of its stroke; the rest of the stroke was completed by the expansion of the steam already admitted. This plan is now used in all engines that are built for economy.

Other inventions made by Watt on his steam engines were: a *parallel motion*, that is, an arrangement of links connecting the end of the piston rod with the beam of the engine in such a way as to guide the rod almost exactly in a straight line; a *throttle valve* for

regulating the rate of admission of steam; and a *centrifugal governor*, which controlled the speed of the engine shaft by acting on the throttle valve. Watt's engine as finally developed is shown in Fig. 4.

Watt saw that by using high-pressure steam he could get more work from it; but as it was not possible to make a very reliable

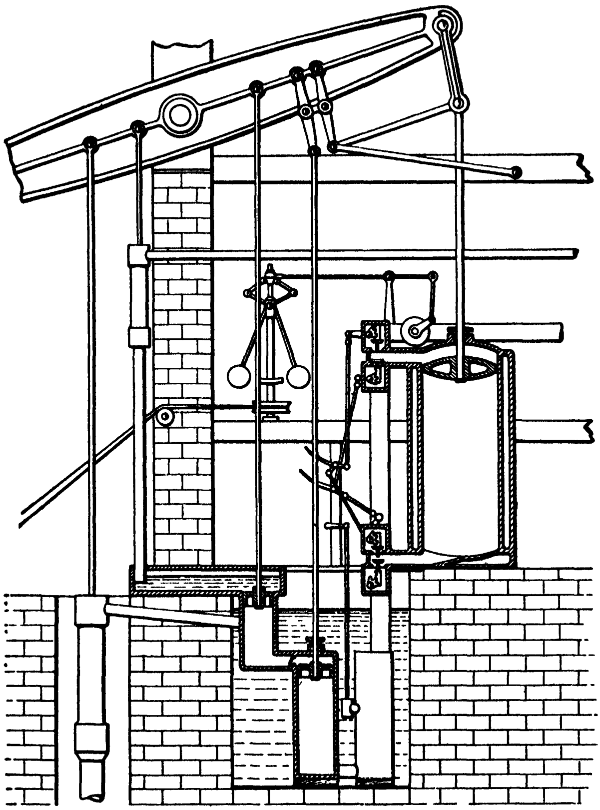


Fig 4 Final Form of Watt's Steam Pumping Engine

boiler he never used a pressure of more than seven pounds per square inch above the atmosphere. About the year 1800, comparatively high pressures came more into use and the non-condensing engine was introduced. In Watt's engine, and all those preceding his, a vacuum was produced in front of the piston by condensing the steam, and either the atmosphere or steam at atmospheric pressure pushed

it through the stroke. In the non-condensing engine, using high-pressure steam, the space in front of the piston could be opened to the atmosphere at exhaust and, although the atmospheric pressure resisted its motion, the pressure of the steam behind the piston was still greater than that of the air. These engines were much more simple than the condensing engines, as they required no condenser.

Compound Pumping Engine. About this time what would now be called a compound engine was introduced by Hornblower and later by Woolf. It had two cylinders of different size, steam being admitted into the smaller one and then passing over into the larger. Only a little expansion occurred in the small cylinder and much more in the larger one.

About the year 1814, Woolf introduced a compound pumping engine in the mines of Cornwall, but a simpler engine was later introduced and Woolf's engine fell into disuse. This later engine became known as the *Cornish pumping engine* and was famous for many years because of its economy. It was the first engine ever built that could compare at all with modern engines in the matter of steam consumption. It consisted of a single cylinder placed under one end of a beam from the other end of which hung a heavy rod which operated a pump at the foot of the shaft. Steam was admitted to the upper side of the piston for a short portion of the stroke and allowed to expand for the remainder of the stroke. This forced the piston down, lifted the heavy pump rod, and filled the pumps with water. Then communication was established between the upper and under side of the piston, exhaust occurred, and the heavy pump rod fell, lifting the piston and forcing the water out of the pumps. Steam was cut off at about three-tenths stroke, and the pump made about seven or eight complete strokes per minute with a short pause at the end of each stroke to allow the valves to close easily and the pumps to fill with water. These engines needed great care and were in charge of competent men, to whom prizes were frequently given for the best efficiency, which doubtless accounts for their wonderful performance.

Parts of Steam Engine. Leaving the historical side of the steam engine let us now turn to the modern simple steam engine and study briefly its construction. Figs. 5, 6, and 7, will serve to illustrate a

horizontal, center crank engine, all the more important parts being numbered. The function of the various parts will be considered in detail later in the work.

Referring to the numbers in Figs. 5, 6, and 7, the names of the parts are shown in the following list:

LIST OF PARTS

Sub-base *1*
Frame *2*
Main bearing caps *3*
Main bearing liners *4*
Cylinder *5*
Cylinder head *6*
False head cover *7*
Valve chest head (head end) *8*
Valve chest head (crank end) *9*
Piston *10*
Piston rings *11*
Piston rod *12*
Piston rod nut (piston end) *13*
Piston rod nut (crosshead end) *14*
Piston rod stuffing box *15*
Piston rod gland *16*
Crosshead *17*
Crosshead shoes *18*
Crosshead adjusting screws *19*
Crosshead pin *20*
Crosshead pin nut *21*
Cross pin washer *22*
Connecting rod *23*
Connecting rod bolts *24*
Connecting rod strap *25*
Crosshead pin box *26*
Crosshead pin box wedge *27*
Adjusting screws *28*
Crank pin box *29*
Crank pin box wedge *30*
Adjusting screws *31*
Crank disks *32*
Crank shaft *33*
Flywheels *34*
Valve pistons *35*
Valve rings *36*
Valve cages *37*
Valve rod *38*
Valve rod nuts (valve end) *39*
Valve rod nuts (ram end) *40*
Valve rod gland *41*
Ram box *42*
Ram box caps *43*
Ram *44*
Ram pin and nut *45*
Ram pin cap *46*
Eccentric rod connection *47*
Eccentric rod *48*
Eccentric rod nut (ram end) *49*
Eccentric rod nut (eccentric end) *50*
Eccentric *51*
Eccentric strap *52*
Dash plate *53*
Dash plate gland *54*
Doors *55*
Door handle *56*
Door clamps *57*
Oil hood *58*
Oil hood handles *59*
Eccentric oil boat *60*
Valve rod oil boat *61*
Oil vent *62*
Sheet steel lagging *63*
Drain cocks *64*
Shaft governor *65*

Sub-Base. The sub-base *1*, Fig. 6, is made of a good grade of cast iron and is usually heavily ribbed and made high enough to permit the wheels to clear the floor. The sub-base is often omitted with engines of large size, the engine being set upon a concrete base.

Frame. The frame *2* is the element or link by which all of the parts of the engine are held in place, so that their relative positions

are always maintained to the end that their proper functions may be performed. The frame is a heavy, substantial casting so designed

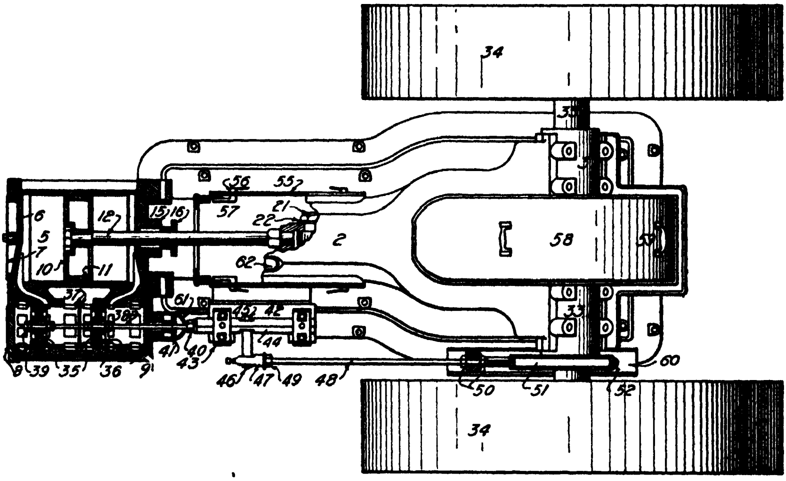


Fig 5 Plan View of Modern Simple Engine

that it is strong enough to take all the stresses put upon it. The type, size, and details of the frame vary with the type and size of

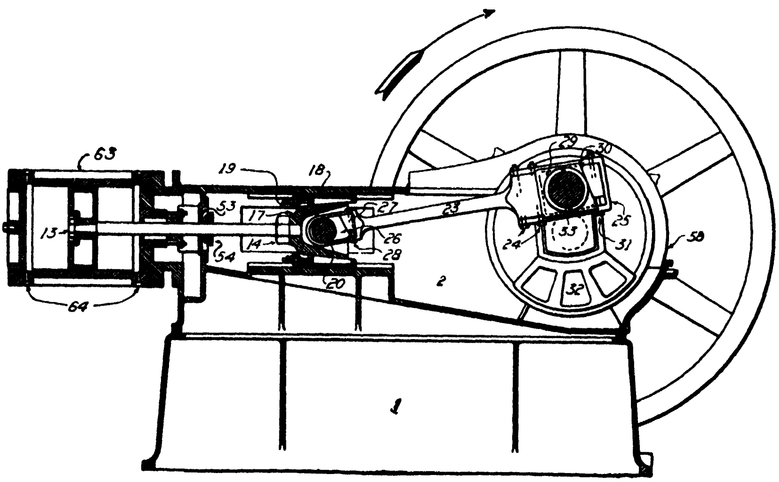


Fig 6. Side Elevation of Modern Simple Engine

the engine of which it is a part. Usually the lower guide, valve rod guide, and seats for the main bearings are cast integral with it. In

small sizes the cylinder is frequently cast integral with the frame. Provision is always made for adjustments necessitated by any wear of the frame or parts attached thereto. It is to be noted in Fig. 6, that the frame crank case *2* is connected with the crosshead guide. It frequently has an opening into the sub-base, thus permitting the oil from the crosshead, guides, and crank to drain into a suitable receptacle in the sub-base, from which it is taken by means of a drain cock conveniently located in the side or end. The crank is enclosed

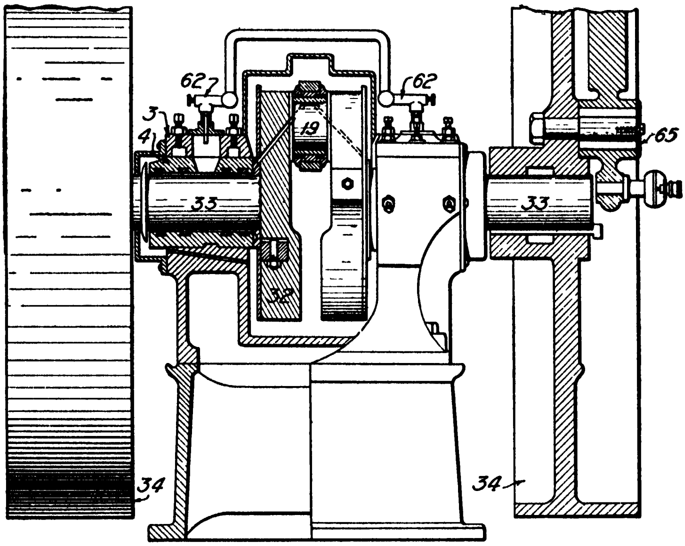


Fig 7. End Elevation and Part Section of Modern Simple Engine

by a neat, pressed, sheet steel cover, which prevents the oil from being thrown outward on the floor while the engine is running. Quite frequently the crank cover is made of cast iron.

Cylinders. The cylinder *5*, Fig. 5, is one of the most important parts of the steam engine, for it is in the cylinder that the energy of the steam is converted into useful work. The cylinder is circular in section and is attached to the bed by means of a number of bolts. It is made of close grain, gray cast iron. The casting of the cylinder should be done with great care, so as to insure a casting free of blow-holes or other defects.

Fig. 8 illustrates the cylinder in cross section as well as showing its contained parts. The cylinder barrel *1* is accurately bored

and fitted. Inside of this barrel the piston *2* is driven back and forth by the steam, which is admitted alternately on one side and then on the other through the ports *13*. The piston is connected to the crosshead through the piston rod *3*. The continuous movement back and forth of the piston causes the surface of the cylinder to wear away, and in order to avoid a shoulder being formed by this action, the cylinder is counterbored at each end by an amount depending on the size of the cylinder. The diameter of the counter-

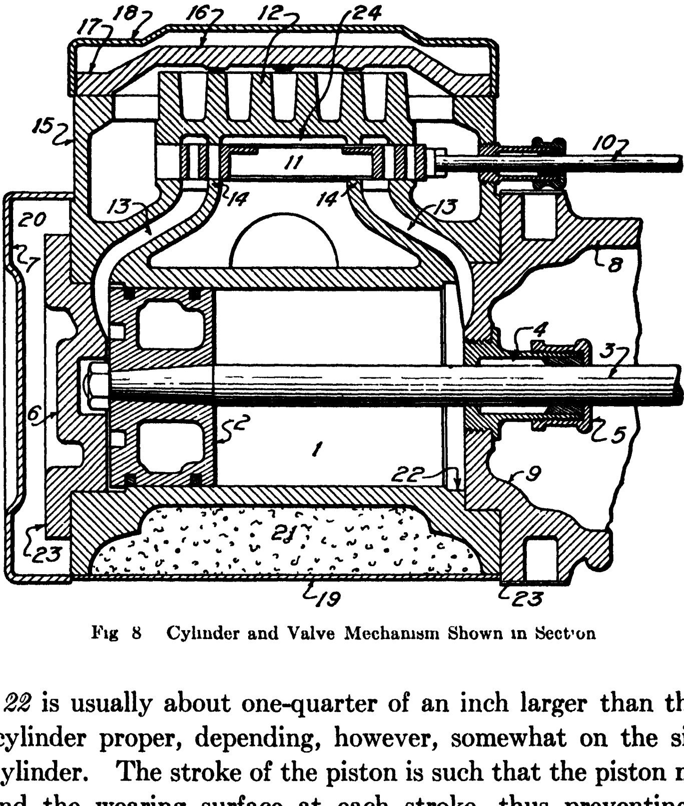


Fig 8 Cylinder and Valve Mechanism Shown in Section

bore *22* is usually about one-quarter of an inch larger than that of the cylinder proper, depending, however, somewhat on the size of the cylinder. The stroke of the piston is such that the piston moves beyond the wearing surface at each stroke, thus preventing any shoulder being developed in the cylinder wall.

The cylinder is attached to the bed of the engine by a number of bolts which are placed through the flanges *23* of the cylinder and cylinder head. Each end of the cylinder is closed by means of the cylinder heads *6* and *9*. The cylinder head *6* is called the back cylin-

der head (head end), and *9* is known as the front cylinder head (crank end). In the illustration the front head *9* is a portion of the frame, but in many constructions it is entirely independent of the frame. In order to have a steam-tight cylinder, it is necessary to make a tight joint between the cylinder heads and the cylinder barrel. This is accomplished by turning both surfaces true, then grinding the joints with emery and oil. After the joints are well ground the heads are tightly drawn up against the cylinder by means of bolts suitably arranged. A sheet iron jacket *19* is put around the cylinder, leaving an air space *21* between the cylinder walls and the jacket. This air space retards the cooling off of the cylinder walls, hence initial condensation of the steam in the cylinder is reduced. In some types of engines such as locomotives, this air space is filled with some non-conducting material such as asbestos. This is also

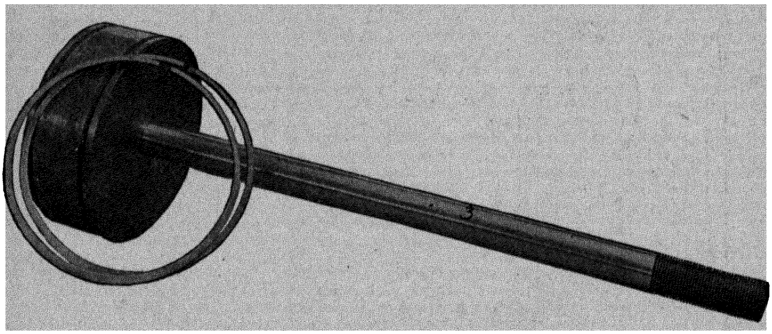


Fig. 9. Piston, Showing Piston Rings for Making Steam-Tight Joints

sometimes done by builders of stationary engines. It should be noted also, that the back cylinder head has an air space for the same reason as that given for the space surrounding the cylinder. Since condensation does take place in the cylinder, some means must be provided for removing the water, hence the drain cocks *64*, Fig. 6, are placed in the bottom of the cylinder at each end. The pipe connection for these cocks enters the cylinder in the counterbore near the wearing surface. Any water that may be in the cylinder will be forced out through these cocks if they are open. Care must be taken that the cylinder is freed of water, for if it is not, on account of the incompressibility of water, the cylinder head may be forced off or other damage result therefrom. Some cylinders are provided

with relief valves, which automatically open when the pressure from any cause reaches a certain amount, thus preventing the bursting of a cylinder head.

Piston Rings. Between the piston *2*, Fig. 8, and the walls of the cylinder there must be a steam-tight joint, so that the live steam can not pass around the piston and be exhausted before expanded, otherwise a great waste of power will be incurred. The requirement is fulfilled by having the piston grooved, as shown in Fig. 9, and fitted with packing rings. These packing rings, commonly called *snap rings,* are turned up slightly larger in diameter than the cylinder and being cut, as shown in Fig. 9, they spring out into the cylinder, always pressing against the walls and forming an almost perfect steam joint. The piston of every engine is made with two or more of these packing rings. The cuts in the rings must not be placed directly in line with each other, otherwise the steam would have a better chance to blow through. In order to prevent this the joints are always placed on opposite sides of the piston. Packing rings are always made of cast iron, and are usually turned up to a uniform section. The outside portion and the two sides are carefully machined.

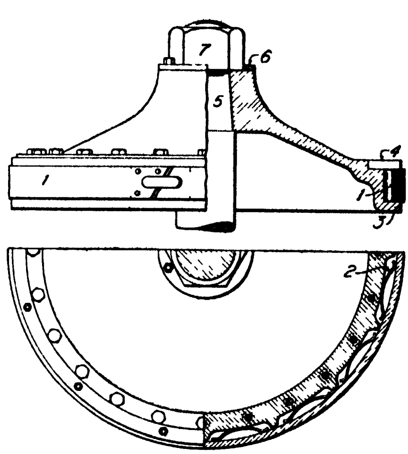


Fig 10 Part Section Plan and Elevation of Conical Piston

Pistons. The piston *2*, Fig. 8, is usually made of cast iron, but sometimes is made of cast steel. It may be a solid disk grooved, as in Fig. 9, or the central portion may be cored out, as in Fig. 8.

Another type of piston that is largely used in marine and sometimes locomotive service is illustrated in Fig. 10. This is a comparatively light cast steel piston, but at the same time a very strong one, due to its conical construction. It will be noted also that only the one packing ring *1* is used. This packing ring is much wider than the ordinary snap ring and is pressed out against the cylinder wall by a number of single leaf springs being placed between the body of the piston and packing ring, as shown in Fig. 10 at *2*. The

piston is made with an L-shaped edge *3*, a band *4* being bolted on the open side of the L, thus forming a groove or opening for the reception of the small springs and the packing ring. The connection of the piston rod to the piston is also clearly shown. The rod *5* has a tapered end which is forced by hydraulic pressure into a tapered hole in the piston; the nut *7* is then tightened up and locked by placing the plate in position around the nut and fastening it with cap screws. This arrangement insures a lasting connection between the piston and the piston rod.

It is essential that the piston be as light as possible in order to reduce the amount of work absorbed in pulling it to and fro, and also to reduce the wear on the lower portion of the cylinder.

The piston rod *3*, Fig. 9, is fitted into a tapered hole in the piston and secured by means of a lock nut and cotter pin placed on the back end. Oftentimes the tapered fit is made very tight and the piston forced on by hydraulic pressure. An older form of attaching the piston rod to the piston is shown in Fig. 8. In this instance the rod has a tapered end, which is driven into a tapered hole in the piston where it is secured by a nut, no cotter pin being used. The other end of the piston rod is threaded, screwed into the crosshead, as shown in Fig. 6, and secured by means of a lock nut. In some constructions the crosshead end of the piston rod is tapered and secured by a key. Many schemes have been employed by different manufacturers for fastening the piston rod to the crosshead, all of which have their advantages and disadvantages. The piston rod, although usually made of a good quality of open hearth steel, is frequently made of nickel steel, which possesses great strength.

Stuffing Box and Packing. As the piston rod passes through the front cylinder head, some provision must be made for making a steam-tight joint between the piston rod and the cylinder head. This is accomplished by means of the stuffing box *4*, and the gland *5*, shown in Fig. 8. Some form of packing is placed around the piston rod within the stuffing box *4* and the gland is forced in by means of bolts or a secured cap as shown, thus holding the packing in the box and at the same time crowding the packing tightly against the piston rod.

The piston packing may be made of woven strands of hemp or cotton; or asbestos may be used. To insure lubrication of the rod

this fibrous packing is soaked in oil before being placed in position. In addition to this form of packing there are different compositions of rubber, graphite, cotton, etc., also various kinds of metallic packing in use. A metallic packing is made of material such as babbitt metal, which is a soft alloy of copper, tin, and antimony. This and other compositions are used for metallic packing, and the metal, being comparatively soft, wears away much more rapidly than that of the piston rod. Fig. 11 illustrates one form of packing, known as the U. S. Metallic packing. The principle of operation is as follows: The babbitt metal rings *2*, consisting of three rings cut in half, provide the packing and are the only parts which come in contact with the rod. These rings are forced into the vibrating cup *6* against the rod, and are fed down as wear takes place by the pressure of the steam itself. The spring behind the follower *3* is merely intended to hold the rings and other parts in place when steam is shut off. A ground joint is made between the flat faces of the vibrating cup *6* and the ball joint *4*. There is also a ground joint between the ball joint *4* and the gland *7*. The

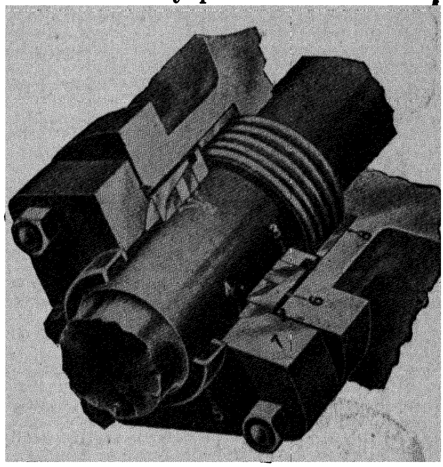


Fig 11 Stuffing Box Packed with Metallic Packing

combination of the sliding face of the vibrating cup and the ball joint permits the packing to follow the rod freely without any increase in friction should it run out of line for any cause. This is an important feature, since the wear of the crosshead, guides, piston head, and cylinder produces an irregular alignment of the piston rod, which would injure the packing to a marked degree, if it was not flexible. The parts of the packing are held in place by the gland *7*, which is bolted to the cylinder head. A steam-tight joint is made between the gland and the cylinder head by means of a copper gasket. The purpose of the swab cup *5* is to hold in place a swab, which is usually made of waste, candle wicking, or a braided material, soaked in oil and oiled from time to time as a means of keeping the piston rod well

lubricated. In addition to this service, the swab catches and retains a considerable amount of dust and grit which would otherwise find its way into the cylinder, where it might do harm. It is to be said in favor of the various so-called rubber packings, that they give very good service. The four different styles of rubber packing illustrated in Fig. 12, are not composed entirely of rubber, but contain other material such as graphite, cotton, etc. These different styles of packing are used both on piston and valve rods.

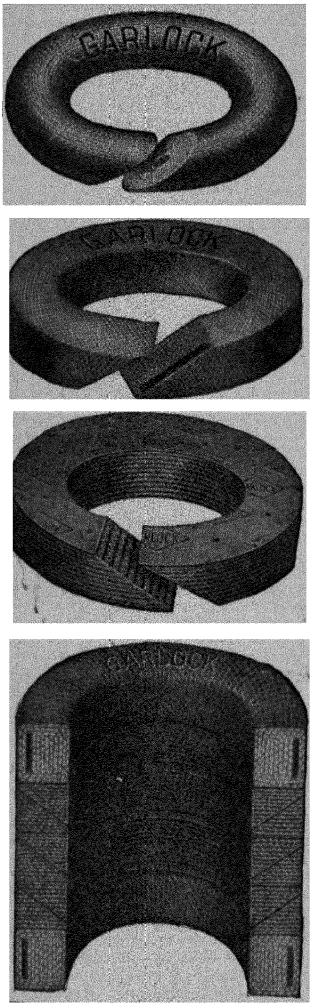


Fig 12 Types of Rubber Packing

It is to be borne in mind that all which has been said with reference to the piston rod is equally applicable to valve stem packing. The general construction of the valve stem glands, vibrating cups, etc., is identical with those of the piston rod. The same materials are used for the packing medium and the same watchful care is required in order to obtain satisfactory results. Packing is an important subject and one which should be carefully looked after. It can not be said that any one particular kind or style of packing is the proper one to use in every case, for a packing which may give very satisfactory results under one set of conditions may utterly fail under another. For instance, a packing suitable for low steam pressures is not efficient where high steam pressures are used, and a packing that may give satisfaction with high pressures may not in any measure meet the requirements imposed upon it by the use of superheated steam. Each particular installation is, therefore, a different problem and must be solved in a different manner.

Valves. In Fig. 8, the valve *11* is shown in position. It will be noted that the valve rests upon the valve seat *14* and works between

the valve seats and the pressure plate *12*. The valve *11* is usually made of cast iron and may be of many different shapes, as will be seen in the study of the various types of engines. There are, however, two general types of valves—one, a plain slide or **D**-valve; and the other some form of piston valve. While there are many modifications and combinations of these two types, yet they are akin to the two types named. The valve in Fig. 8 is of the slide valve type. It is what is known as a double ported valve, that is, steam is admitted to the cylinder by two edges of the valve by reason of the fact that there is an opening through the valve. The pressure plate *12* is used for the purpose of reducing the area of the valve exposed to live steam pressure, it being noted that the portion under the hollow space *24* is not in contact with live steam. This reduc-

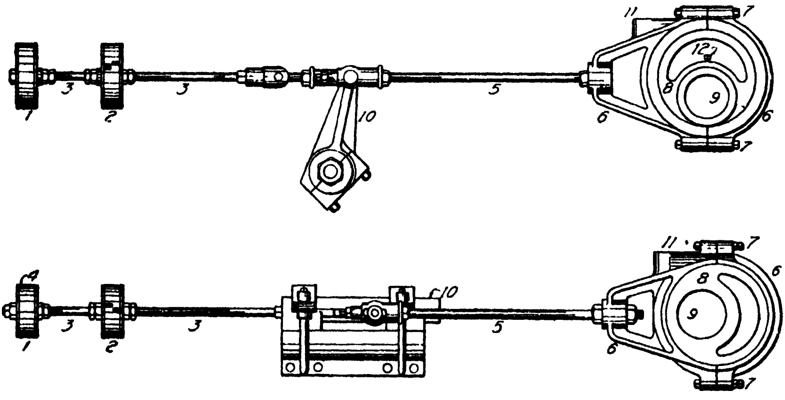


Fig 13 Eccentric Mechanism Showing Rocker and Ram Methods of Connecting Eccentric Rod with Valve Rod

tion of the exposed area is made in order to reduce the amount of effort required to pull the valve back and forth. When the surface of a valve of medium size is considered and an average steam pressure per square inch of 180 pounds is being exerted upon it, some conception can then be had of the amount of friction that must be overcome every time the valve is moved across its seat. To eliminate a portion of this negative work is the primary object of the pressure plate. Pressure plates are of various shapes and designs depending of course upon the type of the engine and the valve used.

One of the advantages the piston valve has over the slide valve is that it is almost perfectly balanced by reason of the fact that steam

surrounds it on all sides, hence there is no excessive pressure on any part of the valve. It is comparatively light and therefore easily driven and lubricated. The general form of construction of piston valves is illustrated in plan in Figs. 21 and 22.

Eccentric. The valve is driven by its connection to the shaft by means of the valve stem, eccentric rod, and the eccentric. The relation of these parts is well illustrated in Fig. 13. The valve shown is an ordinary piston valve with flexible snap packing rings *4* similar to those previously described for the piston packing rings. In fact, the piston valve, as the name implies, behaves very much like the steam engine piston. The two piston ends *1* and *2* are held together by the valve rod *3*. The valve rod has nuts so placed that the pistons are held the proper distance apart. The valve rods, or stems as they are often called, extend beyond the valve box some distance and connect with the eccentric rod *5*. The manner of making the connection between the valve rod and the eccentric rod varies widely, this connection being governed largely by the type of engine and the exigencies of the case. Fig. 13 shows two methods of making this connection, one being accomplished by making use of a rocker arm and the other by using a ram. The way in which the rocker arm *10* is used, is obvious from the figure. The ram *10* is a square block, working in a bearing and so constructed that the valve and eccentric rod can be attached to it. When the ram is used, the motion is transmitted to the valve in a straight line, hence there is less strain upon the connecting parts than if a rocker arm was employed.

The eccentric rod *5*, in both cases, is attached at one end to the eccentric strap *6* and at the other end to the ram or rocker arm. Nuts suitably arranged make the rod secure and at the same time provide a means for lengthening or shortening the rod as needs demand. The valve and eccentric rod are usually made of mild steel turned true and polished.

The eccentric strap *6* is made of gray cast iron, lined with good babbitt metal for a wearing surface upon the eccentric. The strap is held on the eccentric by means of the bolts *7*. By removing liners or shims from between the two sections of the strap, adjustments for wear can be made. There are several patented straps on the market that possess particular features, but the essential elements

of all eccentric straps are about the same. Provision is made for lubrication by having an oil cup *11* cast with the strap.

The eccentric *8* is mounted on the main shaft *9* and is held secure in the position desired by means of the set screw *12*. Eccentrics for large engines are held by means of one or more set screws and a key. For a discussion of the function of the eccentric, the student is referred to the instruction book on "Valve Gears."

Steam Chest. The box *15*, Fig. 8, containing the valve and its parts, is known as the steam chest. The steam chest cover *16* is held in place by studs which pass through the flanges *17* into the box. The steam chest is connected to the steam supply by suitable pipe connections, steam being turned on or off as desired by means of the throttle valve. When the throttle valve is opened, steam passes into the chest through the valve, into the cylinder, where it is

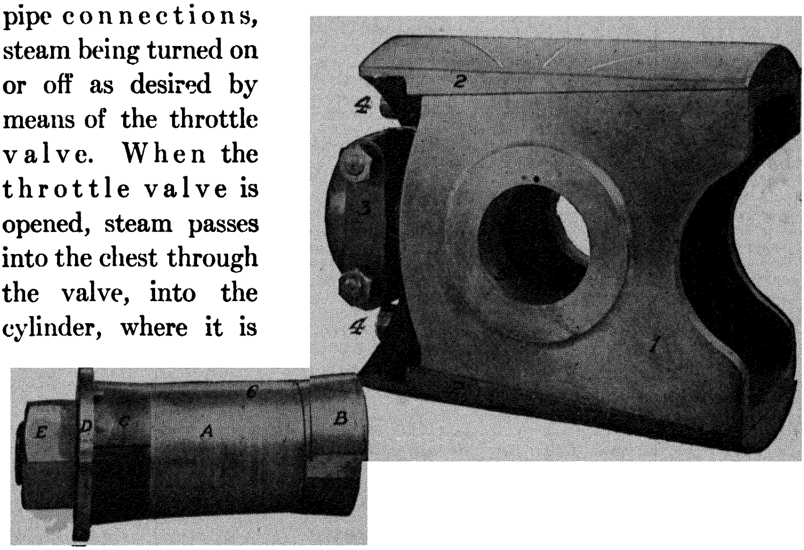


Fig 14 Typical Crosshead and Pin for Large Size Engine

expanded and then ejected through the exhaust opening. The energy of the steam is transmitted through the piston and piston rod to the crosshead *17*, Fig. 6, thence to the connecting rod *23*, crank pin *33*, to the main shaft. In order that these parts may properly perform the function of transmitting this energy, a correct design is highly essential; therefore, a discussion of their construction is deemed necessary.

Crosshead and Connecting Rod. The crosshead is usually made of steel which forms a connecting link between the piston rod and

the connecting rod. It is made in various shapes and patterns. One type is illustrated in Fig. 6. In engines of larger size, the prevailing form of the crosshead used is similar to that illustrated in Fig. 14. This crosshead consists of a steel casting *1* and two wedges, or shoes, *2*, which fit over a projection on the outside surface of *1*. These wedges are either cast or forged and serve as a retainer for a layer of babbitt metal on the outside. It will be noted that there are oil grooves cut on the surface of *2* in order to facilitate the oiling of the crosshead guides. These wedges are provided with a nut and bolt *4*, whereby adjustment for wear can be made as necessary. Usually there is a slight amount of clearance between the crosshead and the guides, but it should not be in any case excessive. The piston rod is fitted into the end *3*, as already described. The connecting rod is attached to the crosshead pin *6*, which fits into the hole *5* and is held in place by a nut. The crosshead pin *6* is made of a good grade of steel and has a portion *B* which fits into the

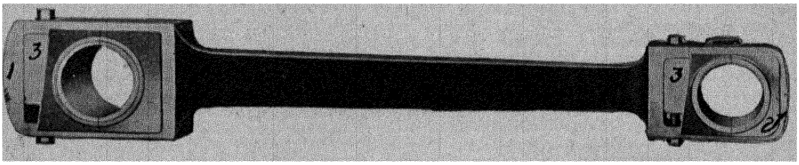


Fig 15. Solid Forged Steel Connecting Rod

back side of the crosshead, as viewed, the other portion *C* fitting into the outside part. When the pin is in place, the collar *D* is adjusted and the nut *E* tightly drawn. The straight portion *A* goes between the sides of the crosshead, and upon it the connecting rod brasses bear.

There are two general types of connecting rods in use, usually classified as marine and locomotive. Connecting rods of the marine type are as a rule used on engines of comparatively short stroke, while those of the locomotive type are employed on engines having a long stroke.

These rods are forged from open-hearth steel, with solid forged ends for the crosshead end, and a square end for the crank end in case of the marine type; and a solid forged or forked end for the crank end in case of the locomotive type

The connecting rod, Fig. 15, is a solid forged steel rod having the ends machined out to receive the brasses. The crank end *1* is

fitted with a brass or bronze box lined with a good quality of babbitt metal. The crosshead end *2* is usually, but not always, fitted in a similar manner to that of the crank end. Adjustment for wear is made by means of wedges at each end, as shown at *3*. These rods are usually of rectangular cross section, although round shapes sometimes are used, especially on small engines.

The marine type of connecting rod is illustrated in Fig. 16. The body of the rod is forged similar to the locomotive type, as is also the small, or crosshead, end, but the distinguishing difference is in the way in which the large, or crank, end is formed. The end of the rod is enlarged and finished square, and the box containing the crank bearing which is lined with a good wearing material, is fastened to the rod proper by means of the bolts. Adjustment for wear is made by tightening up the nuts on the bolts.

It will be seen in Fig. 6 that the connecting rod is the connecting link between the crosshead and the crank *33*. The length of the

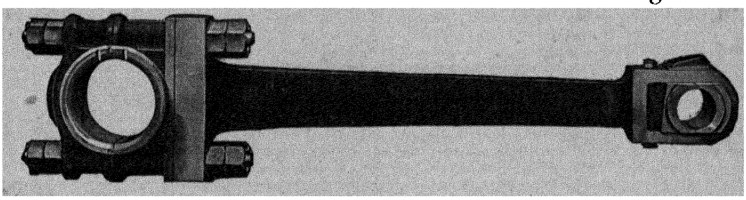


Fig 16 Marine Type of Connecting Rod

connecting rod bears a definite relation to the length of the crank radius. The ratio of the length of the connecting rod to that of the crank radius varies in practice from four to eight. Occasionally conditions demand a greater ratio than eight, but it is seldom less than four.

Fig. 17 illustrates the connection of the piston, crosshead, connecting rod, and crank shaft. The function and construction of the piston, crosshead, and connecting rod have been previously discussed. However, the figure is valuable in that it shows quite clearly the relation of the various parts to each other. The crank shaft used on center-crank engines is frequently a solid steel forging, which includes the crank pin *2*.

Miscellaneous Parts In order to compensate for the weight of the connecting rod and brasses it is necessary to put counterweights on the shaft as shown at *4*, Fig. 17. These counterweights are

usually heavy castings, machined to slip over projections on the crank shaft, and securely fastened thereto by bolts or set screws. The portion of the shaft marked *1*, Fig. 17, fits into the bearings provided for the main shaft or crank shaft, the length of this bearing portion being the distance between the counterweights and the collars *5*. It will be noted that on one end of the shaft is located a disk *3*. Sometimes this disk is forged as a part of the shaft and at other times it is made separate and forced on by hydraulic pressure. The purpose of this disk is usually intended to provide a ready

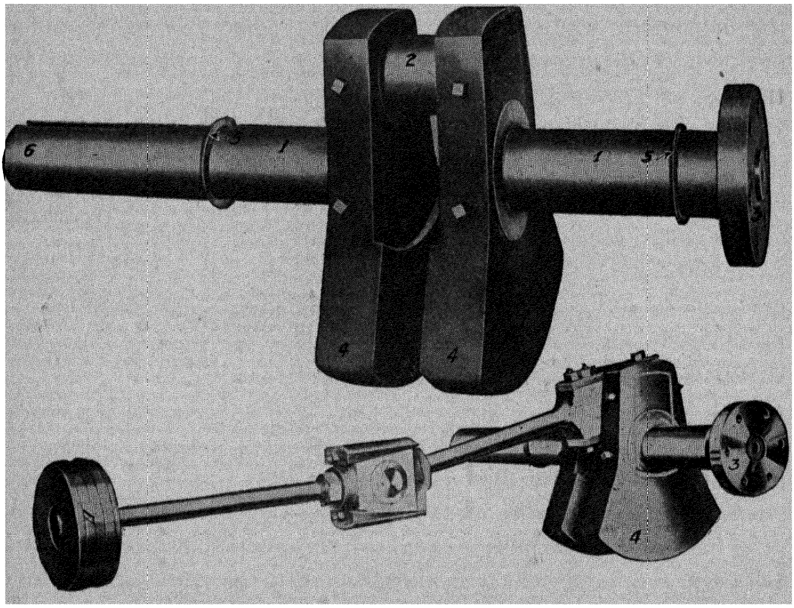


Fig 17 Connection of Piston, Crosshead, Connecting Rod, and Crank Shaft

means of attaching the shaft of an electric generator when a direct connected plant is feasible or desired. It may be said here that a direct connected plant offers many advantages over a belt-driven system. It simplifies the plant, reduces friction, gives greater reliability, and makes possible more power in a given space.

The projection *6* on the other end of the shaft is the axis upon which the flywheel is forced and held secure by means of a key. The crank pin *2* should be of such ample proportions as to be safe against breakage and the heating of the pin or brasses placed upon it.

All engines are not of the center-crank type, but many have a side crank, the crank being a disk or a crank arm fastened on the end of the main shaft very much in the same manner as the disk *3*, Fig. 17. In this kind of construction the crank pin is usually a piece separate from the crank arm or crank disk, and is connected to it by being forced on and then riveted over, or by nuts put on and cottered. In either the side crank or center crank construction, the distance from the center of the axle to the center of the crank pin is equal to one-half the stroke of the engine, as for instance, an 18×24 engine has a crank arm of 12 inches in length, which is just one-half of the length of the stroke. In speaking of the size of the engine it is customary to mention the diameter of the cylinder first, that is, in speaking of an 18×24 engine is meant a cylinder 18 inches in diameter and a stroke 24 inches.

The main bearing *4*, Fig. 7, should be designed with great care, having liberal proportions and lined with anti-friction metal, hammered in place and accurately bored and scraped to fit the shaft. On small engines the lower half of the main bearings are usually made of a part of the frame, the upper half being a removable cap. Between the upper and the lower portion of the bearing, metal liners are placed, which afford ready means for making any necessary adjustments.

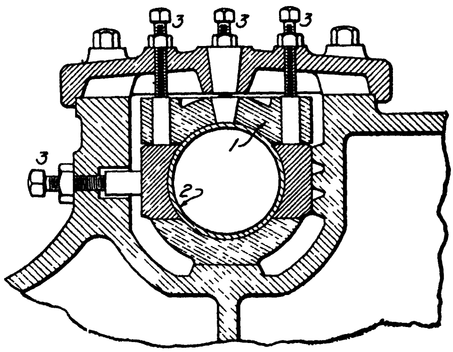


Fig 18 Section of Babbitt Lined, Quarter-Boxed Main Bearing

Large engines have a babbitt lined, quarter-boxed main bearing of ample size, Fig. 18. To provide for both vertical and lateral adjustments it consists of four parts carefully machined on all sides and scraped to fit accurately. This bearing is so constructed that the bottom piece can be removed by slightly raising the shaft. The other three parts are removed after taking off the cap. By use of the adjusting screws *3*, the side *2* and the top *1* may be properly adjusted by the sense of feeling when the engine is in motion.

There are many other types of main bearings besides those mentioned, but they differ only from those already described in some of the minor details. The value of these details varies through wide limits, each builder contending for his own particular design.

A side-crank engine needs but one heavy bearing, such as that shown in Fig. 18, as the flywheel end of the shaft, being subjected to forces acting in but one direction only, requires a much smaller bearing. This outer bearing, Fig. 19, is called an out-board bearing and is smaller and simpler in construction than the main bearing. It is supported by a special casting, which has a hollow recess into which lubricating oil is poured. The shaft carries one or more small chains or rings which fit loosely on the shaft and dip into the

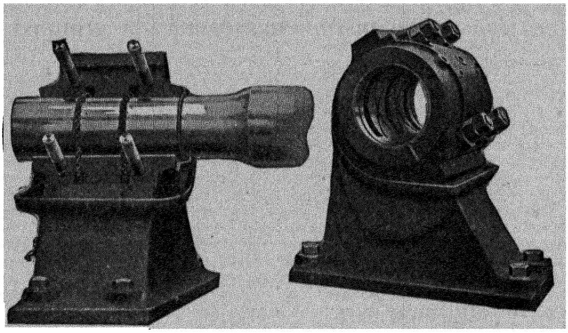


Fig 19 Out-Board Bearing of Simpler Construction than Main Bearing

oil. Thus it is seen that oil is constantly brought in contact with the bearing of the shaft. This same scheme of lubrication is also used for the main bearing. As the different types of engines are considered, the several types of bearings will be noted and discussed.

The belt wheels *34*, Fig. 5, serve a two-fold purpose—one as a governing device, the value of which will be discussed later, and the other as a means of storing up energy while the piston is in mid-stroke, where the crank effort is greater than the resistance to be overcome. The belt wheels act as a flywheel and give up this energy at the ends of the stroke, thus enabling the engine to run over the dead centers. The design of the belt or flywheel is an important item in the proper proportioning of a steam engine. Its weight and dimensions must be very acurately determined. The belt wheel, or flywheel, whichever is employed, is made of cast iron

of various sizes, some being cast solid in one piece, others being cast in two or more sections. In any case the wheel is forced on the shaft and securely fastened thereto by means of a key and set screws.

TYPES AND CONSTRUCTION

Classification. Thus far an effort has been made to give the student some idea of the development of the steam engine and enable him to become familiar with the various parts and their functions. The natural sequence to the above study is to make a detailed study of the several types in use. No hard and fast rule can be given for classifying steam engines as they overlap in so many instances. That is to say, a simple engine may be either condensing or non-condensing; it may be high speed or low speed, etc. According to well-known authorities, various piston engines may be grouped under the following classes:

- I. Number of cylinders
  - Single cylinder
  - Multiple cylinder
- II. Construction of cylinders
  - Fixed cylinder
    - Vertical
    - Horizontal
    - Inclined
  - Movable cylinder
    - Oscillating
    - Rotary
- III. Action of steam
  - Single acting
  - Double acting
- IV. Transmission of steam power
  - Direct acting
  - Indirect acting
    - with balance lever or beam
    - without balance lever or beam

Professor R. H. Thurston in his book entitled "A Manual of the Steam Engine" classifies steam engines according to their purpose and use, as follows:

- I. Stationary mill engines
  - Moderate speed
  - High speed
- II. Agriculture engines
- III. Portable and semi-portable engines
- IV. Road locomotives
- V. Railway locomotives
- VI. Pumping engines
  - Crank and flywheel
  - Direct acting
- VII. Marine engines
  - Paddle engines
  - Screw engines
- VIII. Special types

The same authority further classifies engines according to their structure, as follows:

I. Expansion { Simple; Compound }

II. Position of cylinder { Direct acting; Beam; Vertical; Inverted; Horizontal; Inclined }

III. Steam { Condensing; Non-condensing }

IV. Pressure { High pressure; Low pressure }

V. Piston action { Reciprocating; Vibrating }

VI. Steam turbines

VII. Rotary

VIII. Connection { Direct connected; Geared }

IX. Condensation { Jet condensing; Surface condensing }

They are frequently designated by the name of the inventor, designer, or constructor, as the Watt, the Corliss, or the Porter engine.

From the last two groupings it is evident there is no sharp line of demarcation, for in many instances engines of one class have essential parts similar to those of another type. In this work the classification outlined in the last group will be taken as a basis for study.

Simple Engines. The simplest type of engine is the single expansion. It has one cylinder and admits steam for a part of the stroke, expands it during the remainder, and exhausts either into the atmosphere or into a condenser. Simple engines, Figs. 5 and 6, are now used only for comparatively small powers, say 200 h.p. or less, and although more extravagant in the use of fuel than the others, may still be the most economical financially, if low first cost is an important item; if they are not run continuously; or if the load fluctuates widely.

Compound Engines. Compound engines have two cylinders known as the high pressure and low pressure, Figs. 20 and 21. It will be noted that two different types of compounds are represented, the one in Fig. 20 being known as a cross-compound, the two cylinders

being parallel, and the one in Fig. 21, a tandem-compound engine, the cylinders being in line with each other.

Steam enters the smaller or high pressure cylinder, and then expands until release, when it is exhausted into the larger cylinder, where it expands further. The cylinders should be so proportioned that approximately the same amount of work can be done in each, which may be accomplished by making the high pressure cylinder enough smaller than the low so that when the steam leaves the high at a lower pressure than when it entered it, the increased volume of the steam may be taken care of and at the same time the increased area of the low pressure piston may compensate for the drop in steam pressure.

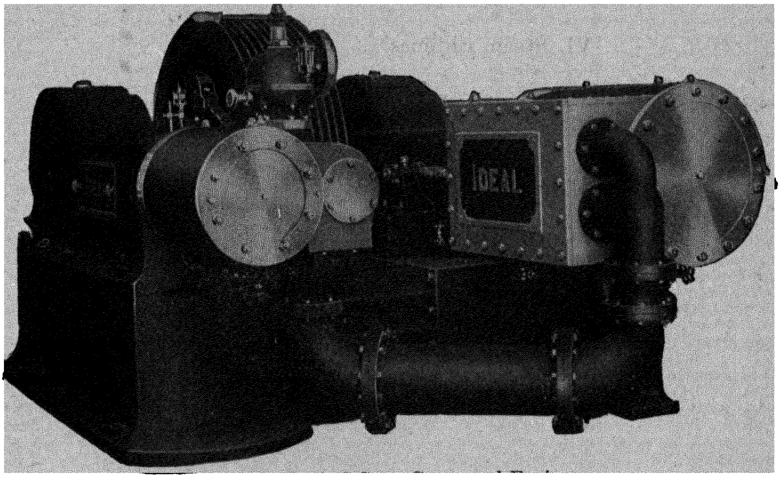


Fig 20 Typical Cross-Compound Engine

Besides being economical, the cross-compound has a distinct mechanical advantage. The two cranks may be set at right angles so that when one is on dead center, the other is at a position of nearly its greatest effort. This makes a dead center impossible, and gives a more uniform turning moment. Then the individual parts may be made lighter and are thus more easily handled.

When the cranks of the cross-compound engine are at 90 degrees with each other the low pressure piston is not ready to receive the steam when the high pressure exhausts; therefore, there must be a receiver to hold the steam until admission occurs in the low. Such engines are called cross-compound, because steam crosses over from

one side to the other. Sometimes instead of having the cranks at 90 degrees, they are placed together or opposite. Then the strokes begin and end together, and the high can exhaust directly into the low without a receiver.

A tandem-compound engine, Fig. 21, has both pistons on one rod, the high pressure piston rod forming the low pressure tail rod. Such engines are less expensive because there is but one set of reciprocating parts instead of two, but like simple engines they have the disadvantage of dead points.

Triple Expansion Engines. Triple expansion engines expand the steam in three stages instead of two. There are usually three

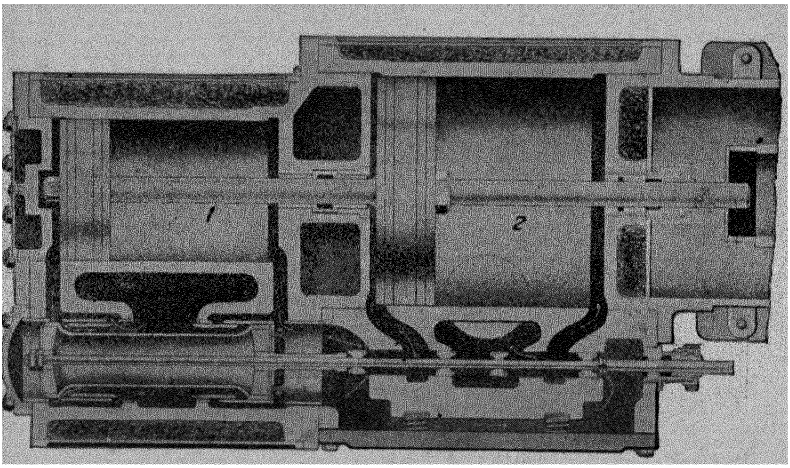


Fig. 21. Section of Cylinder and Valves of a Tandem-Compound Engine

cylinders, viz, the high, the intermediate, and the low, arranged with cranks 120 degrees apart. This gives a more uniform turning moment than a compound. Sometimes there are four cylinders on the triple expansion engine, viz, one high, one intermediate, and two low. This arrangement gives better balance and is often used in marine work.

For triple engines there must be a receiver between each two cylinders. Fig. 22 shows the essential features of a triple expansion engine.

Quadruple Engines. Quadruple engines expand their steam in four stages instead of three. Multiple expansion engines are nearly always condensing.

Cylinder Ratios. There are several considerations to be remembered when proportioning the cylinders of the multiple expansion engines. The ratio of the cylinders should be such that each develops nearly the same power, and the drop in pressure between the cylinders and receivers should be as small as possible.

There are many formulas in use, some simple, others more complex involving mathematical calculation. A common rule for compound engines is to make the ratio of the cylinders equal to the square root of the total ratio of expansion. Thus, if the steam has an expan-

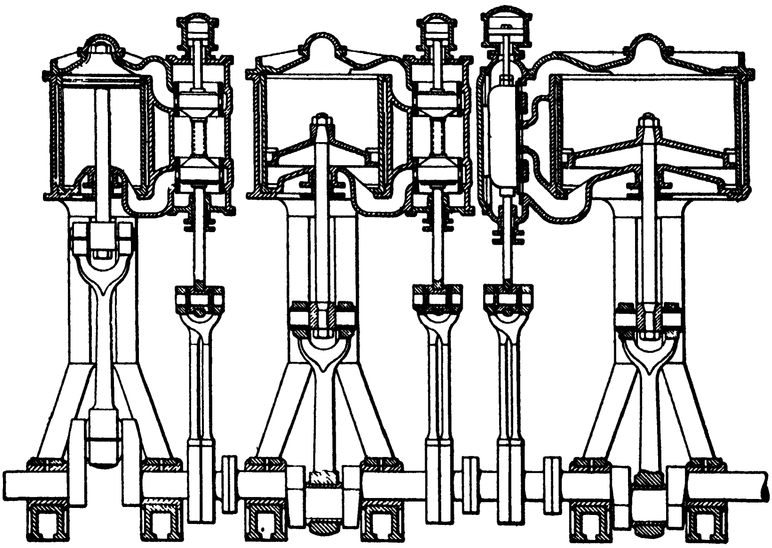


Fig 22 Section of Essential Features of Triple Expansion Engine

sion ratio of 9, the ratio of the cylinder volumes will be $\sqrt{9}$, or 3; that is, the low pressure cylinder will have a volume three times as great as the high pressure cylinder. If the cylinder ratio is 3 and the length of the stroke is the same for both, the diameter of the low pressure cylinder will be 1.75 times that of the high pressure cylinder.

Another rule is to make the cylinder ratio equal to the total ratio of expansion multiplied by the fractional part of the stroke completed when cut-off occurs in the high pressure cylinder.

Suppose the ratio of expansion is 9, as above, and that cut-off occurs at one-third of the stroke in the high pressure cylinder, the ratio of cylinder volumes will be $9 \times \frac{1}{3}$, or 3. If cut-off occurs at one-half of the stroke, the ratio will be $9 \times \frac{1}{2}$, or 4.5.

For triple expansion engines the low pressure cylinder is made large enough to develop the full power if steam at boiler pressure is used.

The intermediate cylinder is made approximately a mean between the high and low. The area of the intermediate piston is found by dividing the area of the low by one and one-tenth times the square root of the ratio of the low to the high.

The above may be written thus:

$$\text{Area of high pressure cylinder} = \frac{\text{Area of low pressure cylinder}}{\text{Cut-off of high pressure} \times \text{ratio of exp.}}$$

$$\text{Area of inter. cyl.} = \frac{\text{Area of low pressure cylinder}}{1.1 \times \sqrt{\text{ratio of low to high}}}$$

In general, for triple expansion the ratios of the volume of the three cylinders are about as follows:

$$V_1 : V_2 : V_3 :: 1 : 2.25 \text{ to } 2.75 : 5 \text{ to } 8$$

For quadruple expansion engines, the ratios are as follows:

$$V_1 : V_2 : V_3 : V_4 :: 1 : 2 \text{ to } 2.33 : 4 \text{ to } 5 : 7 \text{ to } 12$$

It is self-evident that the compound engines illustrated are of the multiple cylinder class. They also have fixed cylinders and are double and direct acting. That is, steam acts on both sides of the piston, and the power is delivered directly from the piston to the shaft or flywheel without the intervention of a walking beam or some other transmitting medium. The engines illustrated in Figs. 20 and 21, are horizontal, whereas the one shown in Fig. 22 is vertical. A horizontal engine is, therefore, an engine whose cylinder is parallel to the ground, and a vertical engine is one which has its cylinder or cylinders perpendicular to the ground. These engines may also be operated either condensing or non-condensing.

From the foregoing it must be obvious that it is not possible to classify an engine within narrow limits, so it appears to be more logical to classify them according to the service for which they are to be used, as in the second grouping.

Selection of Type. In the selection and design of an engine there are a great many factors to be considered. The engine must be as light as possible, and yet must be strong enough to do the work

likely to be imposed upon it. The bearings should be large and ample in number. Lubrication must be given especial attention if high speeds are to be used. Lightness of design tends towards small first cost, which is important, but durability and efficiency should not be entirely sacrificed for low first cost. In the course of time the more expensive engine may prove to be the cheaper as maintenance and repairs may amount to considerable on a poorly designed and built engine. For some classes of service, however, the cheap engine is the one best adapted. For instance, in saw mills, cotton mills, and for similar class of service, a low first cost simple engine is the one best suited for the work, because the labor employed to operate it is often inexperienced and ignorant. In such cases the protection and care that can be given the engine is poor, hence the lower the value of the property exposed, the less will be the loss resulting from the depreciation. On the other hand, if one is selecting an engine for a lighting plant in a city, he would more than likely select one of the most improved types of high speed, condensing machines. In the latter case the first cost would be considerably more than the one selected for the saw mill, but the increased efficiency of operation, the slight depreciation, and the reduction in maintenance would more than compensate for this.

From the foregoing it is evident that there are many factors to be taken into consideration when selecting a steam engine for any given service. In the further study of the several different types the class of service for which each is best suited will be indicated in so far as it is possible to do so. There are, however, some general features every engine should possess independent of its class. It should be simple in construction, having compactness combined with great strength and durability. It should be well balanced and free from severe vibration. Accessibility of parts is also an important consideration.

STATIONARY ENGINES

Simple Side-Crank Type. Stationary engines for ordinary mill service, such as machine shops, small power plants, and various manufacturing concerns, are generally simple engines operating at moderate speed, having either plain slide valves or piston valves. There are, however, some cases where compound engines of moder-

ate size have been installed in similar plants in more recent years. The demand for electric generators has also largely affected the design of steam engines for small electric power plants. In speaking of small plants, in this connection, it may be taken as meaning from 25 to 500 horsepower.

A simple slide valve engine of the side-crank type, which has been largely used in plants where a cheap, efficient engine was the requirement, is illustrated in Fig. 23. This engine has one slide valve, an automatic or shaft governor, and a heavy flywheel which is used as a belt pulley. It is built in sizes varying from 9 inches × 14 inches to 22 inches × 28 inches, and develops a horsepower of

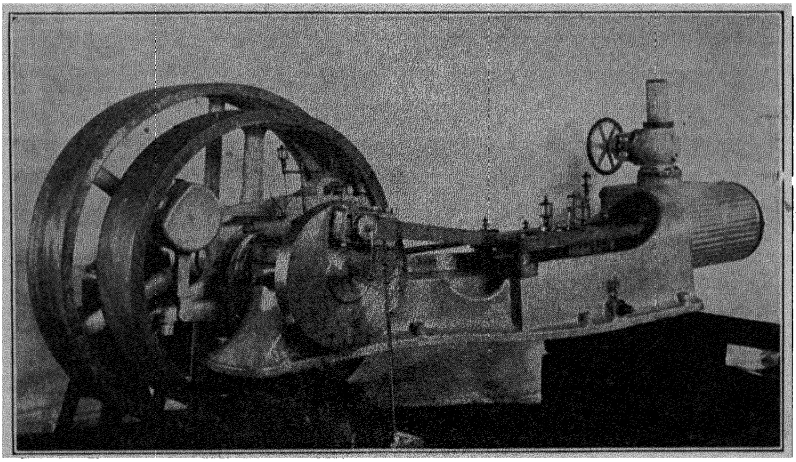


Fig 23 Simple Slide Valve Engine of Side-Crank Type

about 45 to 300 according to size of cylinders, steam pressure used, and the speed at which the engine is operated.

Lubrication of the cylinders is secured by the use of a sight feed lubrication attached to the steam pipe. The main and crosshead bearings are lubricated by oil cups.

This type of engine has been extensively used in cotton gins and saw mills and in small machine shops throughout the country. There are, however, several grades on the market, and it may be purchased for a comparatively low figure where the work to be done does not demand a machine of high grade. This engine has a concrete foundation, is well proportioned, and makes a neat appearance.

Simple Vertical Type. A simple vertical high speed engine that is particularly well adapted for isolated lighting plants in factories, stores, mines, and aboard ships, is illustrated in cross section in Fig. 24. It requires little attention, occupies small floor space, and is not extravagant in the use of steam. The engine is neatly and well designed. It has a large base, which insures stability and rigidity. All of the working parts are enclosed, but readily accessible for inspection and repairs. The frame, cylinders, valves, pistons, etc., are carefully made and adjusted, and the same general types of these various parts conform to the general practice of high speed engines. It will be noted from the illustration that it has a center-crank, automatic governor, and a piston valve. The lubrication of the moving parts is accomplished by means of a geared pump located in the interior of the base of the frame. This pump forces the oil through pipes and grooves to the various bearings. This type of engine is furnished by the makers in sizes from $3\frac{1}{4}$ inches $\times$ 3 inches up to 9 inches $\times$ 7 inches for general service. Much larger vertical engines may be obtained, but are made as a special order. The engine illustrated is so designed to operate at speeds from 250 to 500 revolutions per minute, depending on the size, and uses steam pressure from 60 to 150 pounds. Its commercial rating is from $1\frac{1}{2}$ to 60 horsepower, according to the size of cylinders, steam pressure, and speed of operation.

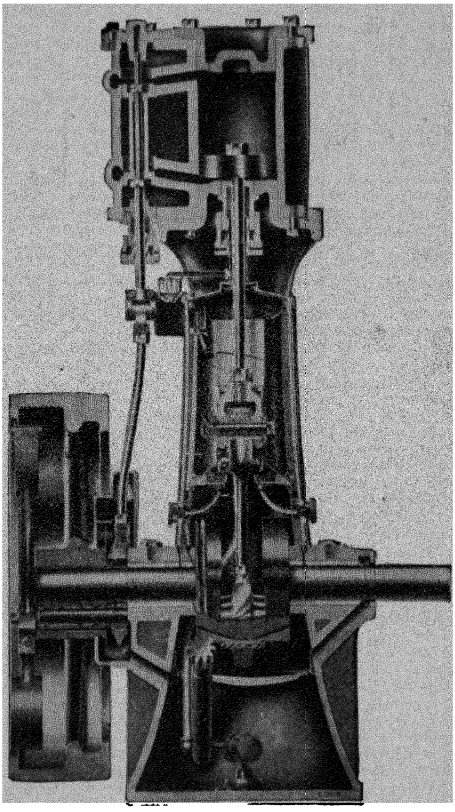


Fig 24 Vertical High Speed Engine

Advantages of Vertical over Horizontal Type. While the discussion given above has had to do with a vertical engine of rather small dimensions and power, yet it must be borne in mind that vertical engines in very large units are built and successfully operated. This leads to a discussion of the relative merits of horizontal and vertical engines. At the present time the most common type of engine is the horizontal direct-acting, that is, an engine whose cylinder is horizontal and whose piston acts on the crank through a piston rod and a connecting rod. In small engines the whole is often on one bed plate. Such engines are said to be self-contained. The cylinder is either bolted to the back of the bed plate or rests directly on it.

In marine work vertical engines are used in almost every case, on account of the *saving of floor space,* which is so important in a vessel. This saving of space is also a very important factor in many other cases, such as in crowded engine rooms in cities where land is expensive.

A second advantage of the vertical over the horizontal engine is the *reduction of the cylinder friction and unequal wear* in the cylinder of the latter. In the horizontal engine the piston is generally supported by resting on the cylinder, which is gradually worn until it is no longer round, causing leakage of steam from one side to the other. This is entirely avoided in the vertical engine.

Still another advantage of the vertical engine is the *greater ease of balancing the moving parts* so that there shall be no jarring or shaking. It is impossible to perfectly balance a steam engine of one or two cylinders. If it is balanced so there is no tendency to shake sidewise it will shake endwise; and if it is balanced endwise it will shake sidewise. The jarring is due to the back and forth motion of the reciprocating parts and the centrifugal force of the crank and the connecting rod. The crank can be readily balanced by making it extend as far on one side of the shaft as it does on the other, but the piston and the connecting rod are more difficult to balance. The effect of jarring can be greatly reduced, if the crank be balanced and the endwise throw made to come in line with the foundation, which should be heavy enough to absorb the vibration transmitted. In a horizontal engine this endwise throw not being in line with the foundation will cause vibration in the engine itself.

In machines that can be anchored down to a massive foundation, a state of defective balance only results in straining the parts and

causing needless wear and friction at the crank-shaft bearings and elsewhere, and in communicating some tremor to the ground. The problem of balancing is much more of consequence in locomotive and marine engines.

To sum up the general advantages of the vertical engines: they

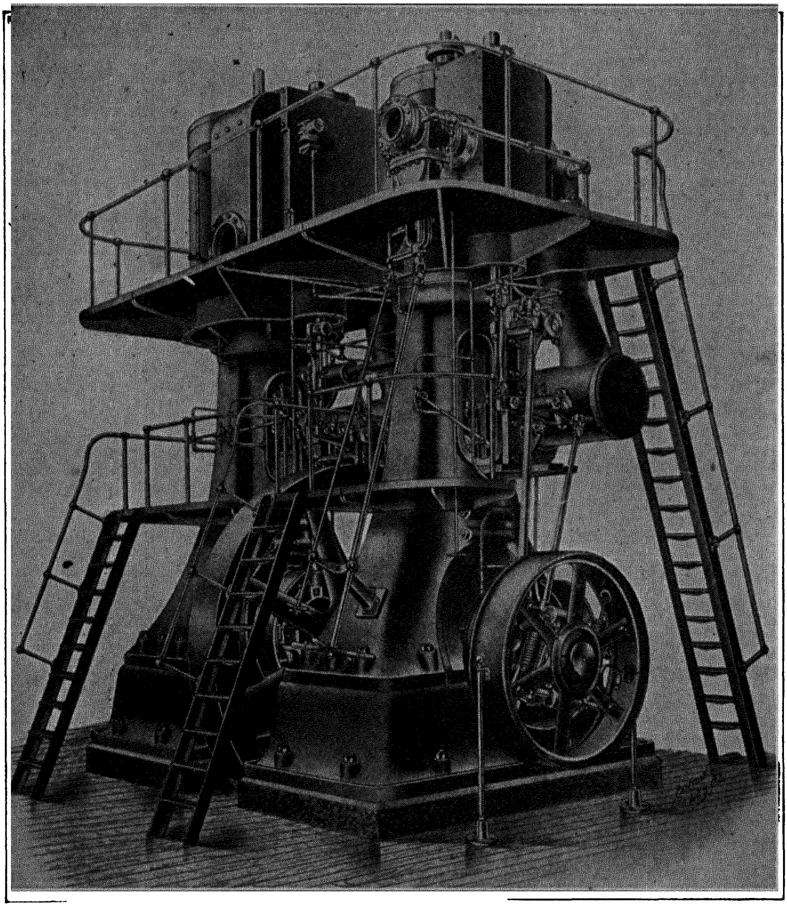


Fig 25 Buckeye Vertical Cross-Compound Engine

have less cylinder wear, they take up less floor space, and they can be better balanced. In addition to these there are certain advantages which vertical engines have for certain kinds of work.

Disadvantages of Vertical Type. The *pressure on the crank pin* is greater during the down stroke than during the up stroke, because

during the down stroke the weight of the reciprocating parts is added to the steam pressure, and during the up stroke this weight is subtracted.

Another difficulty is that in large engines the various parts are on such different levels that they require considerable climbing. This requires *more attendants* and is sometimes the cause for neglect of the engine. The *foundations* for vertical engines need to be deeper than those for horizontal engines, yet they do not need to be as broad.

Buckeye Vertical Cross-Compound Type. The development of electrical machinery and the increased demand for power in con-

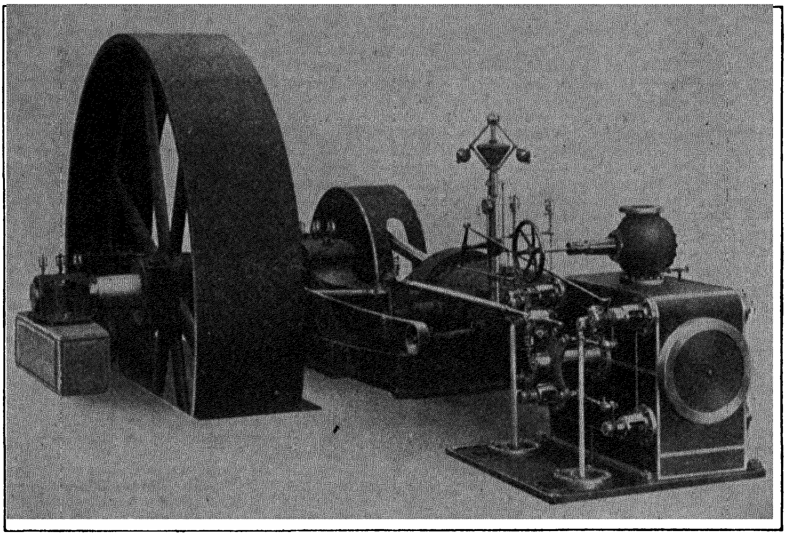


Fig 26 Simple Corliss Engine, Showing Valve Mechanism

gested city locations, where land is very expensive and buildings costly because of their great height, has been the primary cause of the development of large vertical steam engines of various types. The engine, Fig. 25, represents a vertical cross-compound engine as built by the Buckeye Engine Company, which is especially well adapted to electric railway and power and lighting plants, when floor space is limited. The engine may be obtained either as a side or a center crank design. This engine and simple horizontal engines of the same make are typical representatives of economical, high speed engines. They are high priced, but the economy of operation and maintenance make them

very desirable. The vertical engine illustrated may be obtained in sizes developing from 75 to 3,000 horsepower. A discussion of the valve gear used on Buckeye engines is to be found in the instruction paper on "Valve Gears." It is a double valve, giving automatic cut-off as distinguished from throttling cut-off regulation.

Corliss Type. A general utility engine of the highest type both from the standpoint of design and economy of operation and maintenance is the Corliss engine. It is to be found in electric railway power stations; in large and small pumping stations; in blast furnaces and rolling mills; in textile and flour mills; in machine shops and office buildings; in technical schools and colleges; and in nearly all kinds of industrial plants in this country and abroad. The Corliss engine is built in various types, styles, and patterns of any designed capacity up to 10,000 horsepower.

Fig. 26 shows the valve connection and manner of operation of a simple Corliss engine. It will be noted that the engine is governed by a fly-ball governor which is driven by a belt connection to the main shaft. This governor is connected to the steam valves by reach rods. The speed is automatically governed by variation of the point of cut-off. This engine, as well as most engines of this type, has large well-proportioned frames, cylinders, etc. Good workmanship and material enter into its construction, hence it is known as a high-priced engine; but, on the other hand, it is perhaps the most economical in the use of steam.

Valve Mechanism. The distinguishing feature of the Corliss engine is its valve mechanism, a good view of which may be seen in Fig. 27. The gear has four valves, the two top ones being the admission or steam valves and the two lower ones the exhaust valves. There is a connecting rod *7*, Fig. 27, which is connected to the eccentric through a rocker arm, and another rod, as may be seen in Fig. 26. As the shaft revolves, the rod *7*, due to its connection to the eccentric, moves back and forth, and, by reason of its connection through the clamp *8* to the wrist plate *6*, the latter is made to oscillate. The wrist plate *6* is attached to the frame by a pivot projection. The rods *9* have a right and left screw adjustment on each end and transmit motion from the pins *14* on the wrist plate *6* to the steam and exhaust valve bell cranks *10* and *15*, respectively. These valves receive motion in such a manner as to open and close the ports rapidly.

The steam valve bell crank *10* is free to rotate on projections of the bonnet and carries at the end of the lever shown nearly horizontal the brass hook *3* which engages with the catch block. This catch block is rigidly attached to the valve lever *13*, which is keyed to the end of the valve stem, the latter transmitting motion to the valve. Attached to the valve lever *13* is the dashpot piston rod *4*. The hook is so made that it may be automatically tripped when the back part of the hook comes in contact with a cam which is operated by the

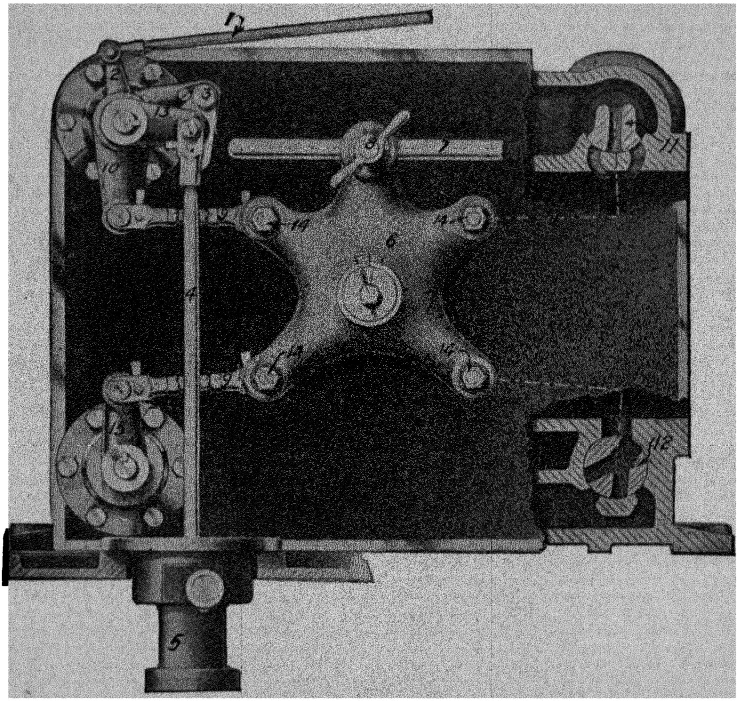


Fig 27 Corliss Valve Mechanism in Detail

arm *2* connected to the governor by the reach rod *1*. The operation of the mechanism is such that the hook may be disengaged at any point of its travel by means of the cam coming in contact with the tripping leg of the hook *3* and causing it to rotate on the pin and move the steel catch out of engagement with the catch block.

The slowing down of the engine, in consequence of reduced steam pressure or an increased load, causes the catch to hold its contact longer and the steam to be admitted longer. In the event that the

speed be increased in consequence of increased steam pressure or diminished load, the hook would be tripped by the cam and the admission valve would be quickly closed by the vacuum dashpot *5*. It must be evident from the foregoing that the regulation obtained by this device must be very sensitive to any change of speed or load. The dashpot *5* closes the steam valve when the hook is tripped by the cam.

The cylinders have four cylindrical holes accurately bored at the four corners, as is shown at *11* and *12* in Fig. 27. Into these openings the valves are placed with their stems and proper packing devices. The seats of the valves are circular. The portion of the valve marked *2* and *1*, Fig. 28, is circular, whereas the remaining portion may have any shape, depending upon the requirements of the design. The valve stem *5-4-6* is also irregular in shape. The portion *4* fits into the slot *3* of the valve and round portions *5* and *6* serve as bearings and as means for attaching the driving mechanism.

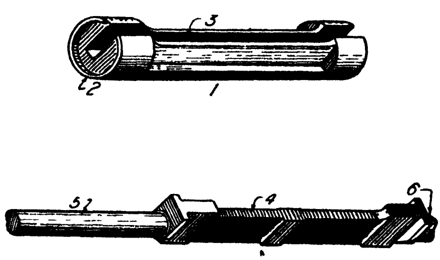


Fig 28. Corliss Valve and Valve Stem

Advantages and Disadvantages of Corliss Type. Perhaps one of the chief disadvantages of the Corliss engine is the large amount of floor space required, a factor which often precludes its use. It possesses many advantages, however, chief among which may be mentioned the rapid and wide opening of the steam and exhaust ports; shortness and directness of ports, which results in small clearance; the adaptation of the steam valve to the functions of cut-off valves; and the location of the exhaust ports at the bottom side of the cylinder, thus draining the cylinders perfectly. Each of these various factors contribute to good engine performance, and their combination has resulted in making the Corliss engine one of the most economical engines manufactured. It will operate upon from sixteen to eighteen pounds of steam per indicated horsepower per hour.

Angle-Compound Type. As an outgrowth of the demand for an engine of high speed and one that will occupy a small space, but which, at the same time, will be economical in the use of steam, there has been developed the angle-compound engine shown in Fig. 29.

Balancing. In an ordinary high speed steam engine, the inertia of the reciprocating parts—namely, the crosshead, piston, and piston rod—and the crosshead end of the connecting rod, is considerable. If a steam engine is to be installed in office buildings, apartment houses, or in other houses where freedom from vibration is a prime requisite, it becomes almost a necessity for the engine to be perfectly balanced. On an ordinary reciprocating engine it is almost impossible to obtain perfect balancing for two reasons:

First, because of the angularity of the connecting rod, which causes the rate of acceleration of the reciprocating parts to be much faster at one end than the other, therefore, the counterweight which exactly balances the forces at one end would be either too light or too heavy at the other end.

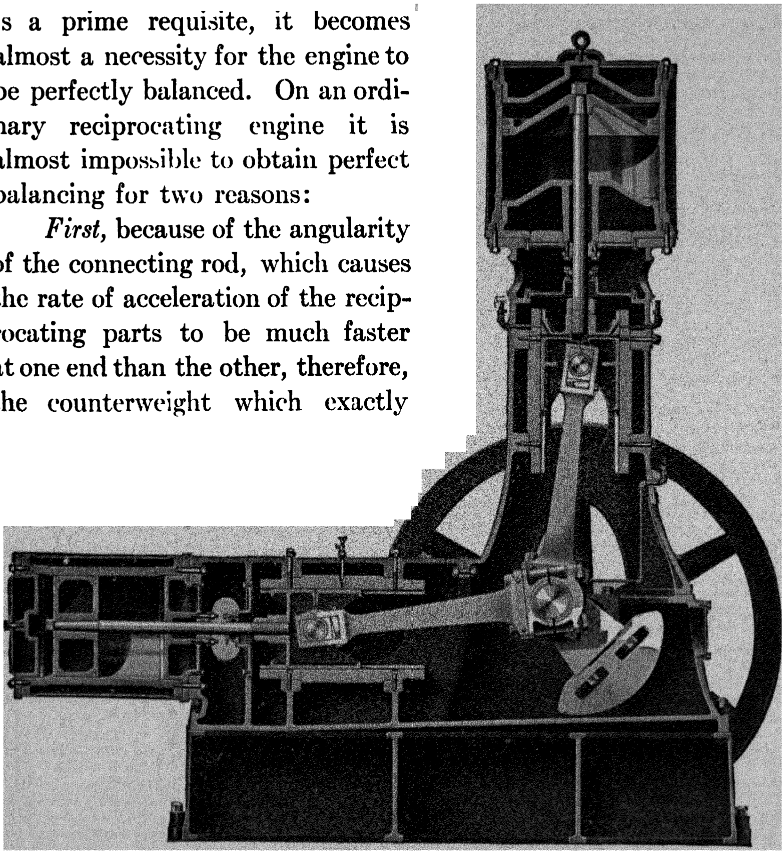


Fig 29 Section of Angle-Compound Engine

Second, the counterweight at all positions in the revolution of the shaft exerts a radial force and when the counterweight is above or below the center of the shaft, there are no reciprocating parts developing a counteracting force, hence the centrifugal force of the

counterweight exerts a powerful unbalanced vertical force. (This has been observed a number of times in locomotive practice where the rails have been bent by the extremely heavy blows of the unbalanced forces.)

In tests at Purdue University on their locomotive testing plant, it was clearly demonstrated that the unbalanced vertical forces are so great at high speeds that the locomotive driver is at times lifted clear off the track. It is obvious from the foregoing that the question of balancing is a serious one, and one that should be carefully considered. A thorough study of the question would involve considerable time and space and the use of higher mathematics.

The several engine builders who put the angle-compound engine upon the market claim for it an elimination of the balancing difficulty. As will be seen from the illustration, the angle-compound consists in combining two engines in such a manner that one crank pin serves both. The high pressure and the low pressure cylinders are placed at 90 degrees from each other in the plane of rotation of the crank. The horizontal engine is arranged so that it is perfectly balanced along its horizontal axis, but is, of course, badly out of balance vertically. On the other hand, the vertical engine is perfectly balanced along its vertical axis, but is out of balance in a horizontal direction. The above statements are true only when we consider each engine separately. When the engines are placed together, the unbalanced effect on one tends to neutralize that of the other. Their relation is such that the same counterbalance serves for both engines. It is claimed for this arrangement that there are four points in the revolution where a perfect balance exists and the resultant effect is to give almost a perfect balance. Another point of interest with these engines is that there are no dead centers; hence by employing a by-pass connecting the two cylinders, the engine can be easily started from any position of the crank.

Summary of Advantages. This type of engine, therefore, possesses the advantage of good balancing; it occupies about one-half of the floor space of a simple engine of the same power; and the compounding reduces its steam consumption considerably below that of a plain slide valve engine.

Uniflow Steam Engine. A type of engine which has a very extensive use in Europe, and which is just beginning to be manu-

factured in this country is the Uniflow engine. Mr. L. J. Todd, in 1886, took out patents in England covering the principle of the Uniflow engine, but Professor Stumpf of Charlottenburg, Germany, deserves credit for developing the engine and for making it a practical success. In Europe the high cost of fuel makes even small economies in the use of steam of considerable value. The Uniflow engine was designed to secure better economy in the use of steam than is possible with other steam engines of equal power. As used in Europe, the poppet type of valve is almost universally employed, because of the better results it gives with superheated steam, which is used practically to the exclusion of saturated steam. The

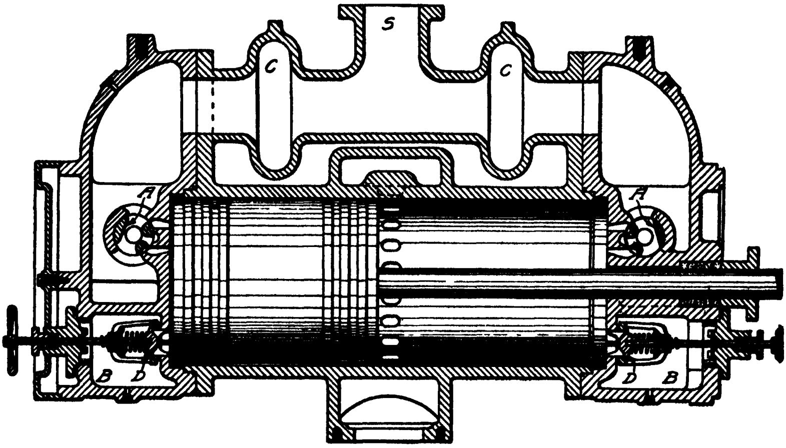


Fig 30 Sectional View of Uniflow Engine
Courtesy of Nordberg Manufacturing Company, Milwaukee, Wisconsin

Uniflow engine is now manufactured in the United States by a few concerns, the Nordberg Company being one of the first to develop and build a successful engine of this type in this country.

Method of Action of Nordberg Engine. A cross-section view of the Uniflow engine as manufactured by the Nordberg Manufacturing Company is shown in Fig. 30. Steam comes to the engine through the inlet S and is led through the passages shown to either admission valve A. These valves are of the Corliss type, not only to conform with American practice, but because, with saturated steam, they give as good results as the poppet type and are less expensive to construct. The steam is exhausted through a ring of ports cast in the middle of the cylinder and is conducted away by the exhaust pipe shown below.

Thus it is seen that the piston performs the duty of an exhaust valve by uncovering and covering these exhaust ports. *D* is a relief valve of large size, communicating with chamber *B*, which is separated by a bridge wall in the cylinder head from the live steam space above. This relief valve serves two purposes: *first,* it relieves the cylinder of any water that may get into it; and *second,* it opens automatically in case the vacuum is lost and prevents the engine from compressing above line pressure. Also, if it is desired to run the engine non-condensing instead of condensing, the relief valve *D* may be backed off of its seat, thus giving the chambers *BB* as the additional clearance volume which is required for non-condensing operation. The drums *CC* on each side of *S* relieve the cylinder and cylinder heads from the strains caused by the expansion of the inlet pipe.

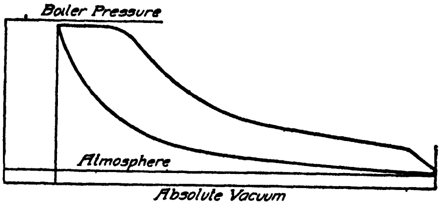


Fig 31 Indicator card for Uniflow Engine Operating Condensing

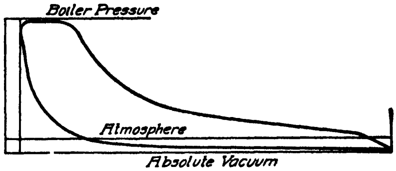


Fig 32 Indicator Card for Uniflow Engine Operating Non-Condensing

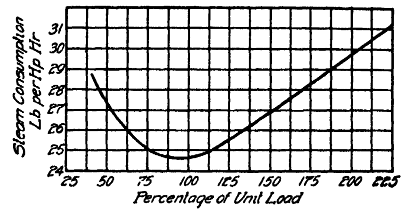


Fig 33 Economy Curve for Uniflow Engine Operating Condensing

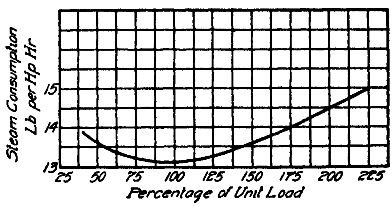


Fig 34 Economy Curve for Uniflow Engine Operating Non-Condensing

Typical Indicator Cards. Typical indicator cards for a Uniflow engine with condensing and non-condensing operation are shown in Figs. 31 and 32, respectively. Figs. 33 and 34 show economy curves when operating, condensing and non-condensing. The cards show the effect of the large exhaust area by the rapid falling off of the pressure as soon as the piston has uncovered the exhaust ports and also the gradual and high compression which is obtained. The economy

curves show that an overload of 100 per cent requires only 10 per cent more steam than for full-load operation when the engine runs condensing, and but 12 per cent more when operating non-condensing.

Chief Factor in High Economy. The chief factor in the high economy of the Uniflow engine is the great reduction of initial condensation. In the ordinary steam engine the piston head and the cylinder head in particular are exposed to the low temperature of the exhaust steam, which cools them considerably and leaves them cooler than the incoming steam. This causes the great loss known as *initial condensation.* In the Uniflow engine the exhaust steam does not pass out near the head of the cylinder, and so does not leave the

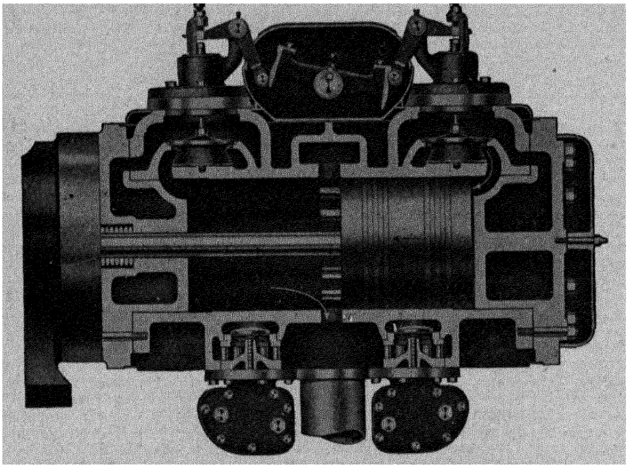


Fig 35 Cylinder and Valve Arrangement in Skinner Unaflow Engine Piston on Head-End Dead Center and Exhaust Taking Place Through Central Ports

cylinder and piston heads cool; and in addition, the compression is carried to line pressure, so that when a fresh supply of steam enters the cylinder it meets surfaces of practically the same temperature as its own. Furthermore, the walls of the cylinder in the Uniflow engine are exposed at each successive point in the stroke to temperatures which are more nearly the same than they are in the usual counter-flow engine; this also helps the engine economy.

Cylinder and Valve Arrangement in Skinner Engine. The form of cylinder and valve arrangement used in the Unaflow engine, built by the Skinner Engine Company, Erie, Pennsylvania, is illustrated in Figs. 35 and 36. It will be noticed that the steam valves

are of the poppet type and are located on the top of the cylinder. Exhaust takes place through central ports in the usual way and also through the auxiliary exhaust valves shown on the bottom side. Fig. 35 shows the piston on head-end dead center with the steam valve at admission and exhaust taking place through the central ports. Fig. 36 shows the central exhaust ports closed, the steam valve on the head end closed, and exhaust taking place through the auxiliary exhaust valve on the crank end.

When the engine is operating non-condensing, the auxiliary exhaust valve for the end in question is opened by the valve-gear mechanism at the point when the central exhaust ports are just

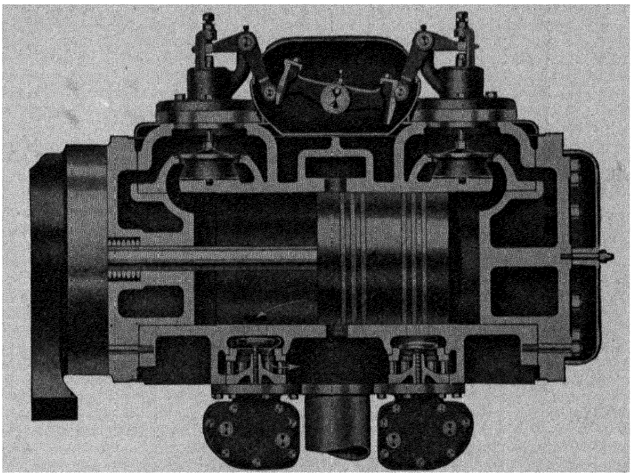


Fig 36 Section of Unaflow Engine Showing Central Exhaust Ports Closed and Exhaust Taking Place Through Auxiliary Exhaust Valve on Crank End

closing, and compression commences at about 35 per cent of the stroke. When operating condensing, the valve-gear mechanism controlling the auxiliary exhaust is automatically disengaged when the vacuum reaches a predetermined amount. Under these conditions the auxiliary exhaust valves remain closed and compression begins at about 90 per cent of the stroke. The construction is such that if when operating condensing the vacuum should fail, the auxiliary exhaust valves are automatically thrown into operation.

American Locomobile. The locomobile is already highly developed in Europe, particularly in Germany, but in this country developments have just recently begun. On account of its high

efficiency, a description of its construction and operation seems desirable.

This apparatus is really a complete power plant all contained in a single unit. It consists of a steam boiler with furnace, superheater, and reheater; a compound steam engine with condenser and vacuum pump; a feed-water heater; and a boiler feed pump. It is now being constructed by different American manufacturers, but that built by the Buckeye Engine Company will be taken as typical, since they were the first builders of locomobiles in the United States.

Details of Power Plant. Referring to Fig. 37, the path of the gases and steam can be traced through the plant, and some of the mechanical features can be seen. Consider first the boiler, furnace, superheater, and reheater. The former is an internally fired, fire-tube boiler, the combustion chamber being at *A* and the fire tubes at *B*. Beyond these tubes is the circular pipe coil *C*, which is the superheater, and still further on is the reheater *D*, consisting of loops of pipe expanded into two headers, as shown. The furnace gases from *A* pass directly through *B*, *C*, and *D*, and then out to the stack through the connection shown in the floor. The boiler is supported on cradle blocks *X*, and the superheater and reheater piping is hung from horizontal beams, as shown.

The steam is led from the dome *E* to the rear end of the superheater and leaves it at the front end, going straight up to the high-pressure cylinder of the engine. Leaving the high-pressure cylinder, the steam is carried to the front end of the reheater and, leaving at the opposite end, is conducted to the low-pressure cylinder of the engine. From the low-pressure cylinder the steam is conducted to a feed-water heater and then to the condenser, neither of which are shown in the figure. In both superheater and reheater, the steam is made to flow against the direction of flow of the furnace gases so that the steam will enter the engine cylinder at a higher temperature.

The engine, as can be seen, is mounted directly on the boiler. The saddle *F*, bolted to the boiler shell, supports the engine bed, which is securely fastened to it. The engine frame is permitted to slide on the saddle *G* so as to allow for unequal expansion between the boiler shell and the engine frame. At the head end of the low-pressure cylinder the yoke *Y Y* supports the cylinder casting.

Fig. 38 shows the relative temperatures of the steam and the

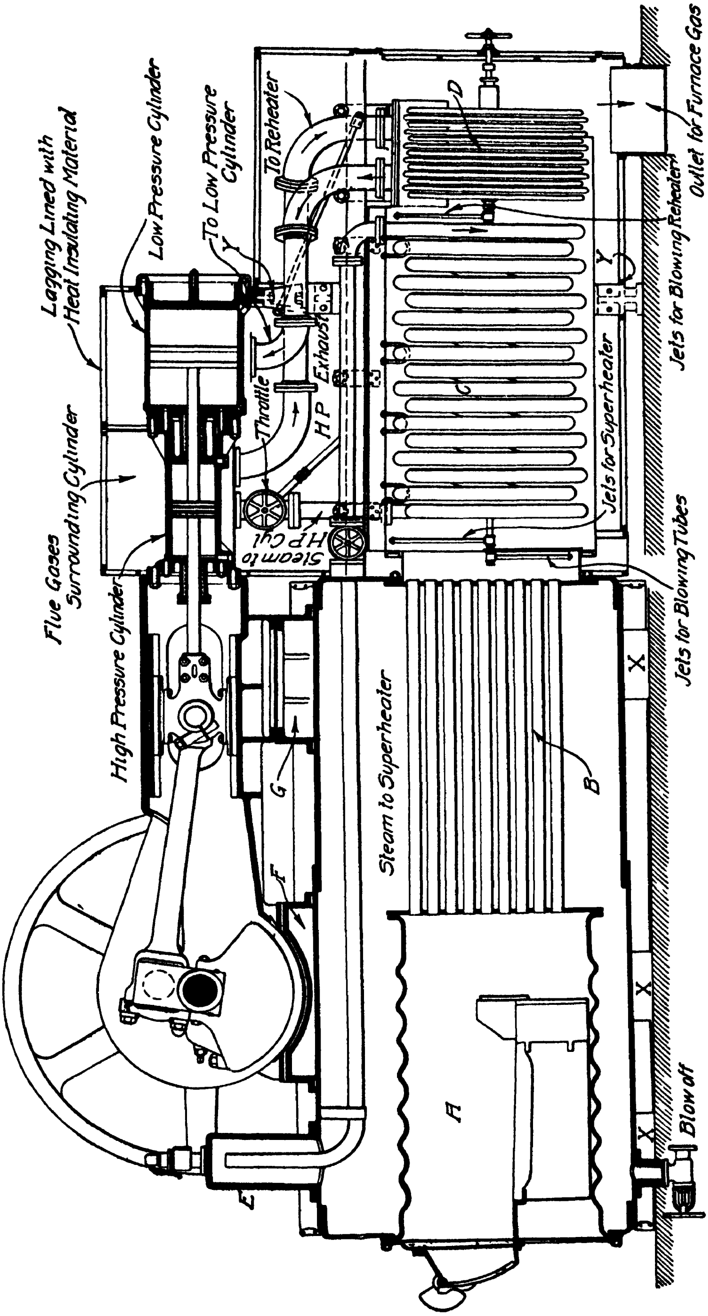


Fig 37 Sectional View of First American Locomobile
Courtesy of Buckeye Engine Company, Salem, Ohio

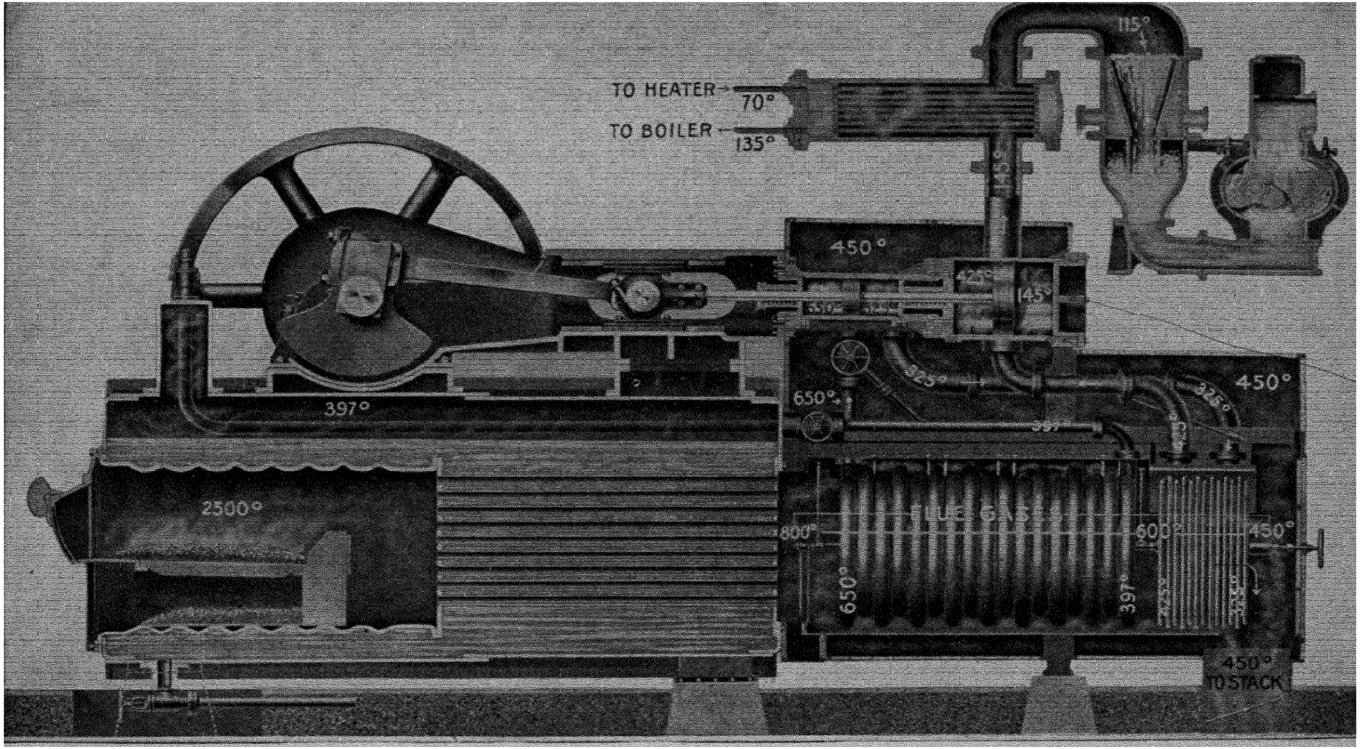


Fig 38 Section of Buckeyemobile Showing Temperatures in Different Parts of the Power Units
Courtesy of Buckeye Engine Company, Salem, Ohio

gas at various points in their respective paths. The figure is self-explanatory, except that the feed-water heater, condenser, and vacuum pump shown in the upper right-hand corner are put there merely for convenience of illustration. In the actual locomobile they are situated alongside the boiler in some convenient manner. The temperatures here shown are approximate and would vary, of course, with different coals, combustion rates, steam pressures, etc. Fig. 38 also clearly shows the manner in which the waste gases circulate around the high- and low-pressure cylinders.

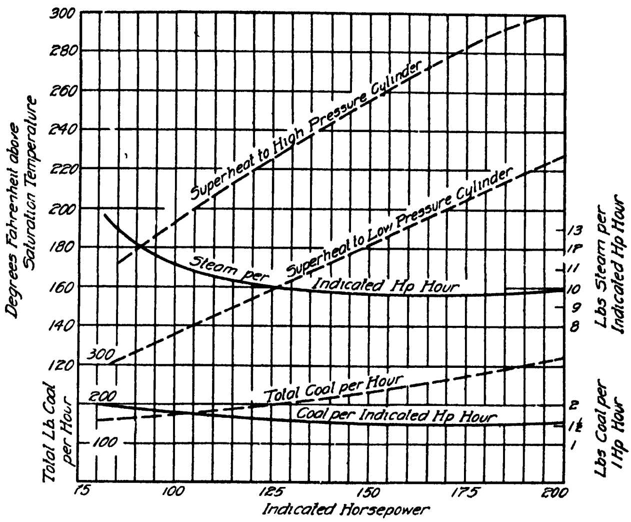


Fig 39 Graphical Results of Locomobile Economy Tests

The big saving aimed at in the locomobile is the reduction of radiation and condensation losses to a minimum. This would mean of course that a greater amount of work could be generated in the engine by the burning of the same amount of coal. In tests conducted in this country, the locomobile has generated an indicated horsepower on less than two pounds of coal, and has used approximately ten pounds of water. Fig. 39 shows graphically the results of tests, which are remarkable considering the size of the unit.

General Survey of Stationary Types. The treatment of the subject of mill or stationary engines, in so far as the scope of this

work will permit, has been covered by the discussions given concerning the plain slide valve engine, the vertical engine of small units, the vertical engines for large installations, the compound and tandem engines which are being used more and more, and finally by a consideration of the most economical engine of all, the Corliss, whether operated simple or compound. In addition to the types mentioned there are a large number of other makes, which have distinguishing features and which give good service, but yet the principles enumerated in the types already discussed fulfill all the requirements likely to be made upon stationary plants. Hence a discussion of other makes is not thought necessary.

FARM OR TRACTION ENGINE

The advancement of scientific and progressive farming has made the farm engine of more interest and importance than ever before; in fact, the demands of the active farmers in recent years have taxed the builders of such equipment to the limit of their output. The steam engine is used for a large variety of purposes upon the modern large farm, and appears most commonly in the form of the so-called traction engine. It is used for plowing, digging ditches, building of roadways, logging purposes, running threshers, and numerous other purposes. Various types of stationary engines of small power are also to be found in use on the farm, the small gas engines now having been perfected to such a degree that they are rapidly replacing the steam engine.

General Description. The traction engine is really more than simply an engine; in fact, it is a self-contained power plant. It consists of a simple or compound engine, a boiler for supplying the steam required by the engine, and the transmission mechanism, together with all the auxiliaries necessary for a complete power plant. A good type of a general utility traction engine is shown in Fig. 40. It consists of a boiler of the locomotive type, carried by four wheels, the two front ones serving as a means for guiding, and the two rear being the ones which receive the power and known as the driving wheels. In order to prevent the slipping of the rear wheels when doing heavy hauling, they are made with heavy projecting lugs or cleats which are forced into the ground by the weight of the machine. The engine, which is mounted on the side of the boiler, as may be

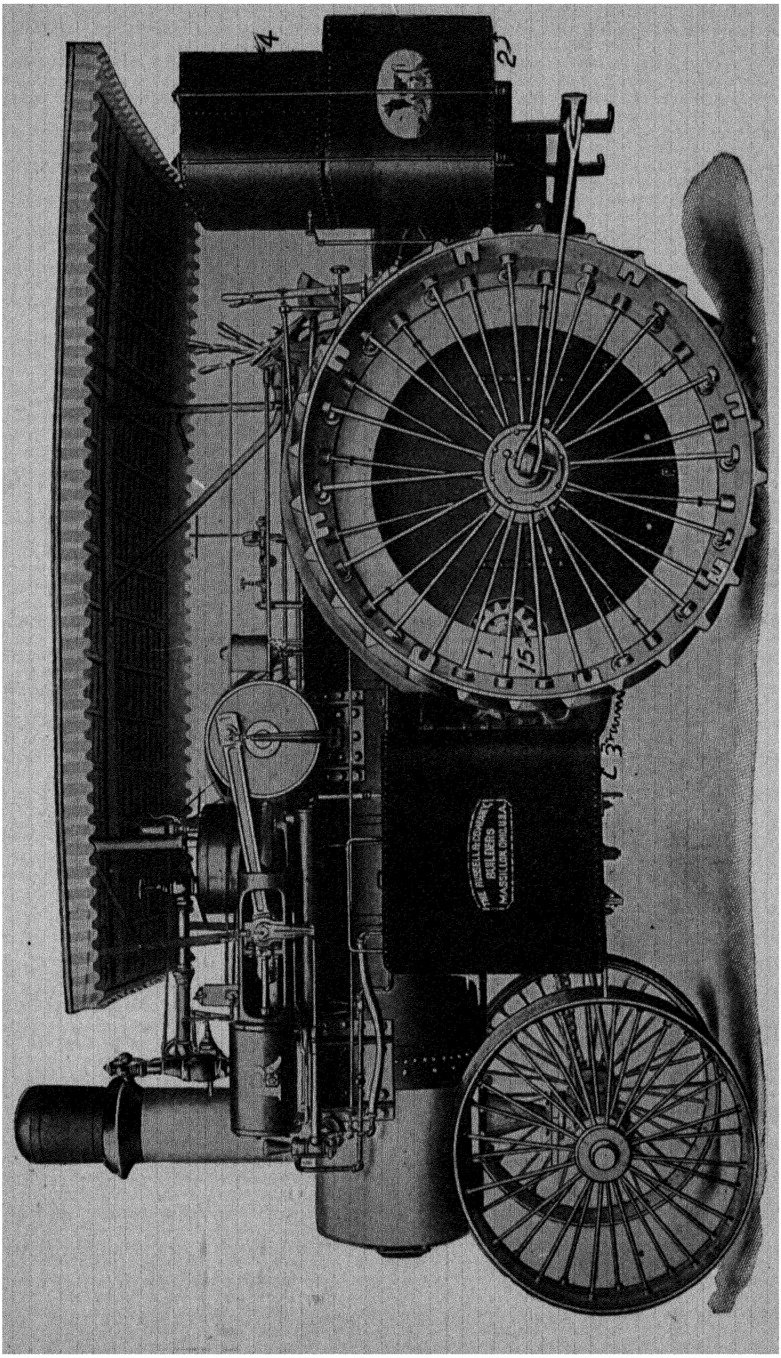


Fig 40 Standard Type of Steam Traction Engine

seen in the illustration, is a plain slide valve engine of the side-crank type. The speed is regulated by an ordinary fly-ball, centrifugal governor. The construction of the various parts of the engine are similar to those previously described in this work. The same watchful care should be given to the lubrication, operation, and maintenance of this engine as to any other, when economy, durability, and reliability are desired. It should be noted that both cross-compound and tandem-compound engines are used as well as simple engines in this class of service, and that various types of valves find application.

In order to make clear the construction and operation of the traction engine, a view showing the rear wheels, platform, and side tanks removed is shown in Fig. 41. The means provided for guiding, reversing, and driving this engine is clearly illustrated. It is evident that the type of boiler used is similar to that of the locomotive boiler, having a narrow fire box. It has an extended front end *1* and stack *2* for carrying away the gases of combustion. The boiler is mounted upon the front wheels through the pivoted pedestal connection *3*. It is supported on the rear wheels, by having the rear axle extend beneath the fire box, or by having the supporting elements riveted to the side sheets as in Fig. 41.

Operation of Plant. *Reversing Mechanism.* The operation of the plant is about as follows: If the engineer desires to go forward the mechanism is placed in forward gear by means of the reversing lever *29*, the reversing being accomplished by means of a swinging eccentric, which can be thrown across the shaft at the discretion of the operator. (On some types of traction engines, a reversing link mechanism is used.)

Transmission. Having adjusted the reversing gear in accord with the desired direction, the throttle valve of the engine is opened by moving the lever *30*. The opening of the throttle valve starts the the engine shaft *12*, which carries the flywheel *11*. On the engine shaft behind the flywheel is keyed a small spur gear which is in mesh with the larger gear *13*, which in turn meshes with the gear *14*. As the engine shaft revolves, the small gear in the shaft revolves, which transmits its motion to *13* and on to the small gear *15*, which is keyed to the shaft driven by the wheel *14*. The gear wheel *15*, Figs. 40 and 41, is in mesh with an annular gear on the drive wheel *1*, Fig.

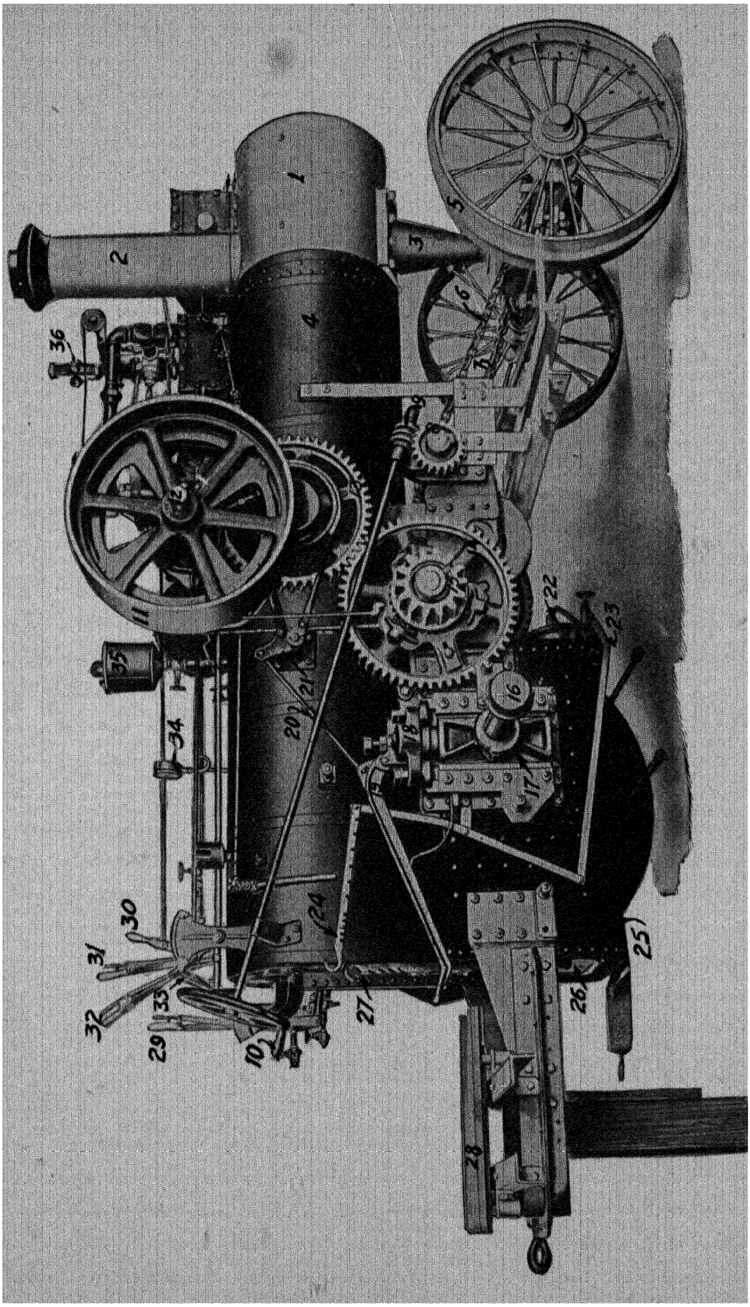


Fig. 41. Traction Engine with Rear Wheels, Platform and Side Tanks Removed to Show Steering and Driving Mechanism

40; hence, by reason of this connection, the large wheel is made to revolve. The shaft carrying gear wheel *15* extends beneath the boiler to the opposite side and drives a set of gear wheels which causes the other driving wheel to revolve with the one just considered.

Running Gear. The axle *16* of the wheel has a sliding head *17* attached to it. This head is free to move up and down in guides securely fastened to the fire box. This sliding head works against a spring, which is contained in the box *18*. This spring reduces the shocks to which the machine is subjected when on the road, hence, the engine is much easier to ride than it otherwise would be. In addition to the easy riding qualities, it also relieves the parts of the machine of stresses and strains due to sudden jolts, which would be detrimental to the durability of the machine as a whole.

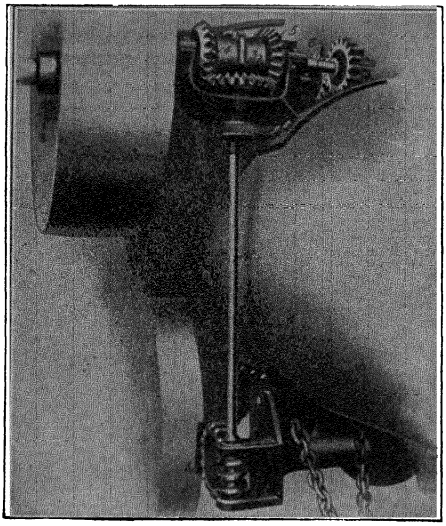


Fig 42 Friction Gear Device for Steering Traction Engine

Steering Gear. The engine is guided by the hand wheel *10*. It will be noted that a chain is connected to the front axle on either side of the pivotal point. This chain wraps around a cam arrangement on the shaft which carries the small gear wheel *8*. The wheel *8* is in mesh with the worm *9*, which may be turned by the hand wheel *10*. If the driver when moving forward should wish to turn to the right, for instance, he would turn the hand wheel so the wheel *8* would be driven counter-clockwise, and in so doing the chain *6* would be shortened, the chain *7* lengthened, the wheel *5* would be cut in, and the machine would turn to the right. If it was desired to go in the opposite direction, the reverse operation would be carried out, that is, gear *8* would be revolved clockwise.

It is sometimes difficult to operate the steering gear by hand, especially in large traction engines and in places where a heavy load

is being driven over rough ground; hence, some engines are provided with a friction gear device. This attachment, Fig. 42, is exceedingly simple, and when it is used the engine furnishes power to guide itself. It consists of a shaft *2* extending from the worm gear *1* to a bracket on the side of the boiler in front of the main shaft. On top of this vertical shaft is a horizontal miter gear *3* arranged to engage alternately with two vertical gears, one at the right *5* and the other at the left *4*. These vertical gears are on a shaft run by a chain of small gearing *6* from the engine shaft. They are thrown in or out, at the pleasure of the operator, by means of a shifting yoke which is worked by a straight rod extending back to the right-hand side of the engineer. A lever at the end of this rod is within easy reach all the time. By moving it forward or backward, the engine is guided to the right or left, as desired. If the lever remains at the center, the engine guides straight. An extension rod is placed on the rear end connecting with a hand lever at the left side of the platform, so that the engine may be guided equally well from either side. To operate this steering lever requires no appreciable exertion on the part of the engineer.

Friction Clutch. A friction clutch is provided in the flywheel, which permits the engine to be operated without driving the machine forward on the road. With the engine running at full speed, the clutch can be gradually thrown into action, and the machine will start forward on the road without any sudden shocks. The clutch is operated by the lever *31*, Fig. 41. By disconnecting the engine from the flywheel, a high speed can be obtained, so that by throwing the clutch in gear quickly the engine is often able to pull the machine out of difficult places. Oftentimes it is desired to operate the engine independently of the traction wheels for the purpose of running the thresher, saw mill, electric generator, or for other purposes, hence some form of clutch is necessary.

Brake. A friction brake is operated by a system of levers and rods as *19*, *20*, and *21*, Fig. 41. The operator can apply the brake by pushing downward upon the foot piece on the lever *19*. The amount of air admitted to the fire box is controlled by the two dampers *22* and *26*, which may be manipulated by the levers *24* and *27*.

Water Tanks. In Fig. 40, large tanks *2*, *3*, and *4*, are shown. These tanks are water reservoirs from which the supply pumps take water and deliver it to the boiler. Opposite the tank *2* is a bin for

holding the fuel, which may be wood, coal, or straw, depending upon the location and character of work to be done. If the traction engine is used for threshing purposes, it would have a fire box arranged for burning straw; whereas if it was being used in a logging camp or a saw mill, the available fuel would be wood, hence the fire box would be constructed accordingly.

Boiler. Since it is necessary to have a high-grade, durable, and economical boiler in order to have an efficient and reliable machine, it is thought advisable to call especial attention to the type of boilers used in this connection and point out some of their good and bad features. It was mentioned in the description of the traction engine, Fig. 40, that a locomotive type of boiler with some modifications was used. Fig. 43 illustrates such a boiler. It is of the fire tube horizon-

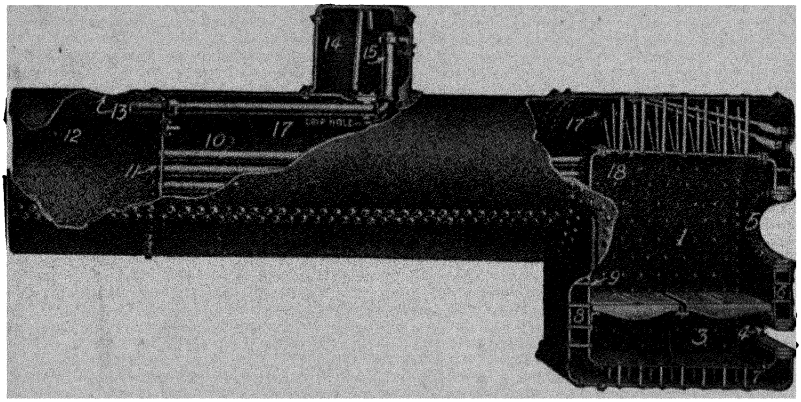


Fig 43 Traction Boiler of the Locomotive Type

tal type. The fire box *1* is of a horizontal rectangular construction with open grate bars *2* and ash pit *3*, below. The fuel, either wood or coal, is fed through the fire door *5*, and the ash is removed through the door *4*. The products of combustion, such as smoke, hot gases, etc., pass through the tubes *10* into the front end *12*, from whence they are exhausted through the opening *13* and the smoke stack into the atmosphere. If straw is to be used as fuel, a brick arch is placed in the fire box which deflects the gases toward the fire door so that, after passing over the arch, they are drawn out through the tubes in the usual manner. It is also necessary to put in different grate bars where straw is used, as the bars must be closer together so the fuel will not drop through into the ash pan.

It will be noted that the fire box is surrounded by water legs *6*, *7*, and *8*, and the water and steam space *17*. Water is also circulating around the tubes and is several inches deep above the crown sheet *18*. As combustion takes place in the fire box and the hot gases pass through the tubes, the plates of the fire box and the tubes become heated. As a consequence, the water in contact with these hot surfaces becomes heated also, and steam is formed which rises to the top of the boiler, entering the steam dome *14* from whence it is taken by the pipe *15* through a throttle valve to the cylinder. By using a steam dome a better quality of steam is obtained, because it is so far above the water level that less water is carried over by the steam into the steam pipe *15*.

This type of boiler has many advantages as well as some disadvantages. It has a large amount of heating surface and it is well distributed. Due to the large amount of heating surface and the excellent draft arrangement, a high evaporation per square foot of heating surface is obtained. It is well adapted to various classes of service and operating conditions. Its disadvantages consist largely in the cost incurred in its maintenance especially in localities where bad water must be used. When this condition is imposed upon it, the flues give trouble by leaking around the joints where they enter the flue sheets *9* and *11*. This leakage may at times become troublesome and in the end costly if proper preventive measures are not taken regularly. Some criticism is also made of this boiler on account of the necessity of using stay bolts in the crown sheet and water legs. It must be admitted that stay bolts are also an item of considerable expense in bad water districts where high steam pressures are used. But by watchful care and manipulation this boiler will give splendid results and for some classes of service it has no equal.

The type of boiler, shown in end view in Fig. 44 and in longitudinal cross section in Fig. 45, is a modification of the well-known and efficient Scotch marine boiler. The boiler consists primarily of a cylindrical fire box *1* enclosed by a circular shell. About midway of the fire box is placed a bridge wall *7*, which deflects the hot gases upward against the shell of the fire box. Ordinary cast-iron grate bars are inserted as at *4*, with the ash pit below. It is to be noted there is a water space *6*, which extends the entire distance around the circular fire box. Above the fire box there are a number of return tubes *3*,

which take the hot gases from the rear end *8* of the boiler to the smoke stack. The path of these gases is indicated by the arrows. To protect the rear sheet from the heat of the gases, a protection plate *9* is riveted or bolted to the plate. As steam is generated it rises, enters the steam dome *12*, passes into the steam pipe *13*, and on to the engine.

It should be noted that this boiler contains no stayed portions and that all the surfaces are circular in form and securely riveted. There being no stayed surfaces the circulation of the water is not interfered with—which is an important consideration—and the opportunity for scale and sediment to collect is greatly reduced, hence there is less likelihood of portions of the boiler becoming heated to the point of injuring the boiler or impairing its safety. Still another feature of interest in the boiler is that the gases are made to traverse the entire length of the boiler twice before being ejected at the stack. This being the case an opportunity is given for a greater portion of the heat contained in the gases to be absorbed by the water, thus securing a higher thermal efficiency than obtained from boilers of the locomotive type. Having no stayed surfaces and a small number of flues results in a small maintenance cost of this type of boiler.

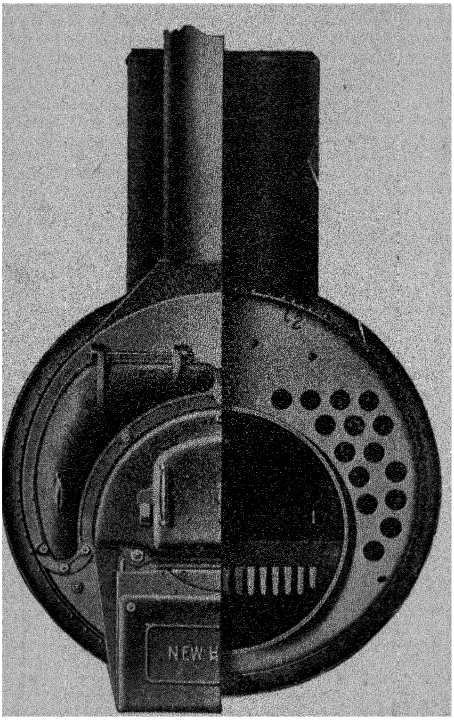


Fig 44. End View of Modified Scotch Marine Boiler

Traction engines run in sizes from about $7\frac{1}{4}$ inches $\times$ 10 inches to 12 inches $\times$ 12 inches for single engines, and for compound engines the common sizes are $5\frac{3}{4}$ inches $\times$ $8\frac{1}{2}$ inches $\times$ 10 inches to $9\frac{1}{4}$ inches $\times$ 13 inches $\times$ 11 inches. The corresponding horsepower

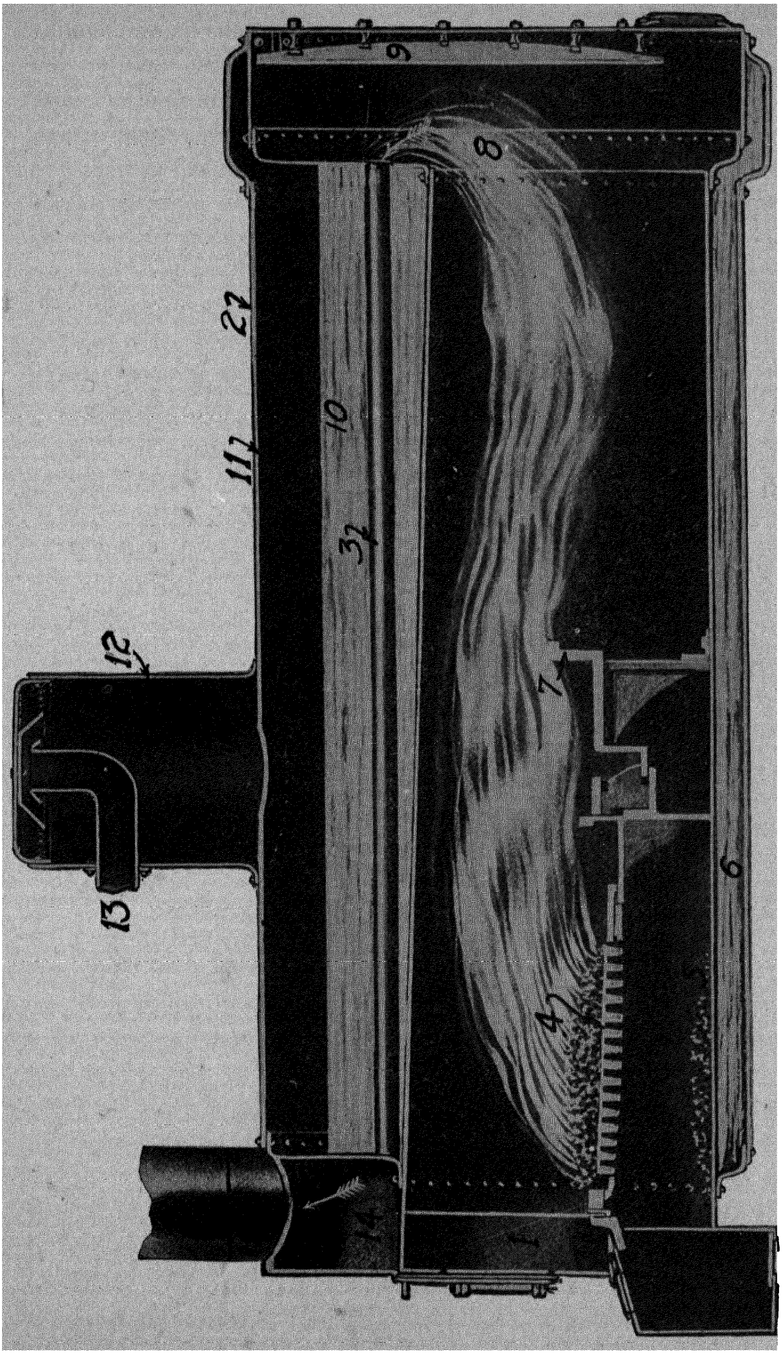


Fig 45 Longitudinal Cross-Section of Modified Scotch Marine Boiler

developed will run between 15 and 100. The speed attained on the road in miles per hour is about $2\frac{1}{2}$ to 5.

Road Roller Type. The traction engine just considered as an agricultural engine may also be considered as a portable engine or a road locomotive. A portable engine is, therefore, one that can be easily moved about from place to place, or as in the case of the traction engine it may be mounted upon wheels and self-propelled.

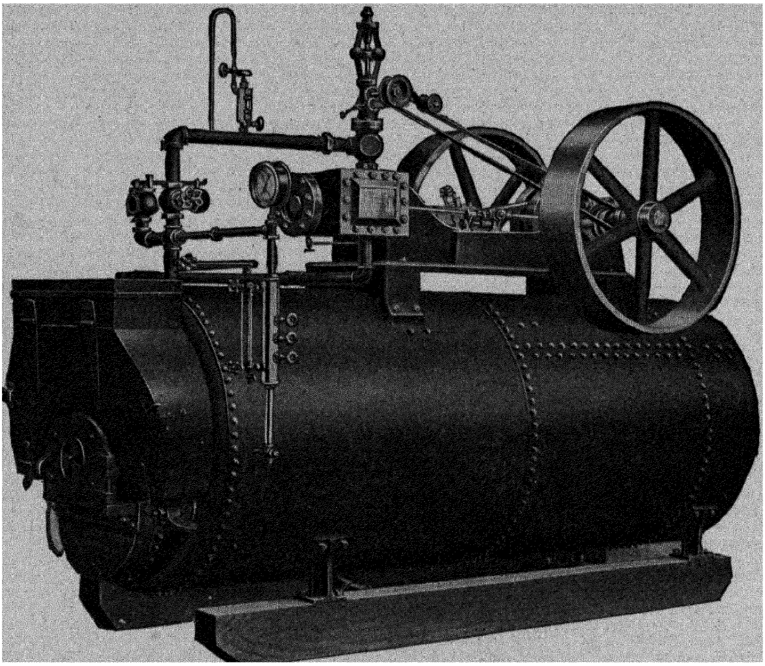


Fig 46 Semi-Portable Engine and Boiler

Another illustration of a similar type is the ordinary road roller, or road locomotive as it is sometimes called. The principle of its construction and operation is similar to the traction engine, the chief difference between the road roller, or road locomotive, and the ordinary traction engine being that the two front wheels of the traction engine are replaced by a large smooth roller, or cylindrical weight, which revolves as the engine moves. The drive wheels of the road roller are also made large, heavy, and contain no cleats or lugs. These rollers are used in the making of macadamized or other forms of roads.

Semi-Portable Type. The semi-portable engine is usually connected with a small boiler and the two together may be moved from place to place as required. It is not mounted upon wheels but rather on large wood skids, and is moved by being placed either in a wagon or on rollers. It is largely used for hoisting purposes in connection with the construction of large buildings, bridges, etc.

Since the portable and semi-portable engines have no transmission mechanism, they are lighter and considerably cheaper to construct than traction engines.

A very neat, compact, and serviceable type of semi-portable engine is illustrated in Fig. 46. It is mounted upon skids so that it may be easily moved about. The engine is mounted on top of a Scotch marine boiler, similar to the boiler last described, and is of the plain slide valve, center-crank type, with a centrifugal governor. The boiler is equipped with a pressure gauge, water glass, and such other appliances as are usually found in a boiler room of moderate size. The boiler used is sometimes of the locomotive type and, oftentimes, both engine and boiler are of the vertical type. The smaller units are usually of the vertical type, the larger ones of the horizontal type. The semi-portable plant is built in sizes ranging from about 20 to 70 horsepower. If the semi-portable plant, Fig. 46, be mounted on wheels and drawn by horses or some other means, then it is usually classed as a portable engine as distinguished from a semi-portable or traction engine.

LOCOMOTIVE ENGINES

It is not within the province of this work to fully discuss the modern railway locomotive, but suffice it to say that no other power-developing unit has been so rapidly developed with such economical results. Considering the exacting demands made upon a locomotive, its performance is remarkable. The locomotive consists of two primary elements, namely, the boiler which generates the steam and the engines which convert the energy of this steam into useful work by giving motion to the transmission mechanism.

Boiler. Fig. 47 illustrates a modern locomotive boiler. It consists of a cylindrical barrel and an enlarged rear end which contains the fire box. The fire box is securely fastened in the boiler shell by stay bolts and radial stays. A few rows of sling stays are sometimes

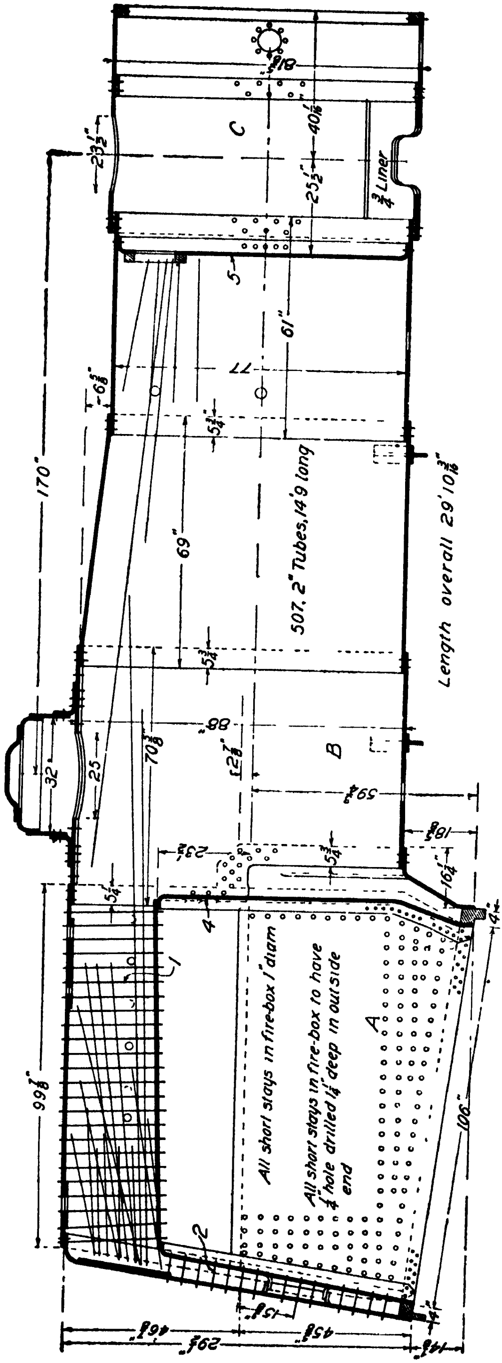


Fig 47 Detailed Drawing of Modern Locomotive Boiler

used at the front end of the fire box to allow for expansion and contraction of the sheets. The boiler is divided into three distinct departments, as the fire box *A*, the water space *B*, and the smoke box *C*. The sheets *4* and *5*, which separate these departments, are known as the back and front flue sheets, respectively. The flue sheets are drilled with holes to receive the flues.

Flues. In the particular boiler illustrated about 400 2-inch flues are used. These flues extend from flue sheet to flue sheet and form a passage for the gases to travel from the fire box to the smoke box. Surrounding the flues in the space *B* and surrounding the fire box is water, which is vaporized into steam due to the combustion of fuel in the fire box. The total amount of heating surface will vary from 2,500 to over 4,000 square feet, according to the type and size of the locomotive. Of this total amount of heating surface only a very small per cent is furnished by the fire box, there being usually only about 200 square feet of heating surface contained in the fire box. It is evident, therefore, that the flues are a very important part of the locomotive boiler.

Grate Area. The amount of grate area varies from about 40 to 60 square feet. It must be obvious that in order for so small a grate area to supply sufficient heat to such a large amount of heating surface there must be a very high rate of coal consumption per square foot of grate area. A series of tests made at St. Louis during the Exposition in 1904 demonstrated that the amount of dry coal fired per square foot of grate area per hour varied from 20 to as high as 130 pounds. These results were obtained from several different types of locomotives operated under widely different speeds and loads, hence the above figures may be taken as approximating the maximum and minimum consumption under ordinary running conditions. Under these very widely different operating conditions it was found that the equivalent evaporation per pound of dry coal varied from $6\frac{1}{2}$ to 12 pounds, which compares very favorably with stationary boiler performance which gives an average evaporation of about 8 pounds of water per pound of coal.

Mechanical Efficiency. The mechanical efficiency of a locomotive is also very good. Through a long series of tests conducted on a well-equipped locomotive testing plant, a mechanical efficiency of 65 to 85 per cent was obtained. The same degree of efficiency has

been obtained in various other tests and under more adverse conditions. The locomotive is also very efficient in the use of steam. The St. Louis tests showed that simple freight locomotives gave an average minimum water consumption per indicated horsepower per hour of 23.67 pounds. The water consumption per indicated horsepower per hour under maximum load was 23.83 pounds, whereas the maximum rate was 28.95 pounds. For compound freight locomotives the average steam consumption was: minimum load 20.26 pounds, maximum load 22.03 pounds, and maximum consumption 25.31 pounds. The average steam consumption for simple passenger locomotives was: minimum load 18 86 pounds, maximum load 21.39 pounds, and maximum consumption 24.41 pounds. When these figures are compared with those of the best stationary engines, some idea of the economy of the locomotive can be obtained. The steam consumption of an automatic, tandem-compound, condensing stationary engine with piston valves under full load is about 18 pounds per indicated horsepower per hour, whereas the compound non-condensing locomotive is about 21 pounds. A Corliss engine or a medium speed, four-valve simple engine will give a minimum steam consumption of about 22 pounds per indicated horsepower per hour under full load. A simple freight engine under full load will use about 23.5 pounds of steam per indicated horsepower per hour. The foregoing figures speak well in favor of the economy of a steam locomotive, which is operated under conditions unfavorable to the securing of good economy.

Engine Characteristics. The engines used on locomotives may be simple or compound; in fact, both are used extensively, although the simple type predominates. It is to be noted that the steam locomotive is equipped with two separate and distinct engines—one being attached to each side of the boiler, and both attached to the driving wheels through the medium of the frames, etc.

The mechanical construction of these engines is quite similar to that of the type already described in this work. Certain features are made necessary in order to properly tie together the engine, boiler, and transmission mechanism. Perhaps the most noticeable change in detail is in the construction of the cylinders and valve seats, otherwise there is little variation from the well-established principles of engine design. The valves, rods, crossheads, guides, etc., are

made of the same high-grade material and constructed in the same first-class manner as is required for a good stationary engine. This being true, much of the discussion of the steam stationary engine and its parts already given is applicable to the engines of a locomotive. There are, however, many perplexing questions that arise with reference to the performance and operation of the locomotive as a whole that are never encountered in stationary practice, due to the unusual and sometimes trying conditions under which the locomotive must be operated. The solution of these problems demands a great amount of ingenuity and engineering ability.

To discuss the various types of locomotives and tell the many interesting and important points connected therewith would require entirely too much space, so the discussion must be confined to narrow limits.

Types of Locomotives. There are certain types of locomotives common in American practice which have special names. The eight-wheel or "American" passenger type of locomotive has four coupled driving wheels and a four-wheeled truck in front. The "ten-wheel" type has six coupled drivers and a leading four-wheel truck. This type is used for both freight and passenger service. The "Mogul" type is used altogether for freight purposes; it has six coupled drivers and a two-wheel or pony truck in front. The "Consolidation" type is used for heavy freight service. It has eight coupled drivers and a pony truck in front. There are also a great many special types for special purposes. In switch yards a type of engine is used which has four or six drivers with no truck. The Forney type has four coupled driving wheels under the engine and a four-wheel truck carrying the water tank and fuel. This type is used on elevated roads largely. "Decapod" engines are a type used for heavy freight service, having ten coupled driving wheels and a two-wheel truck in front. A tank engine is one which carries the feed water in tanks on the engine itself instead of in the tender, as in other engines.

A locomotive of modern design that is being largely used for fast freight service and for heavy passenger service is illustrated in Fig. 48. It is commonly known as the Atlantic type locomotive, having four leading truck wheels, four coupled drivers, and a two-wheel trailing truck. The leading truck wheels serve in a guiding

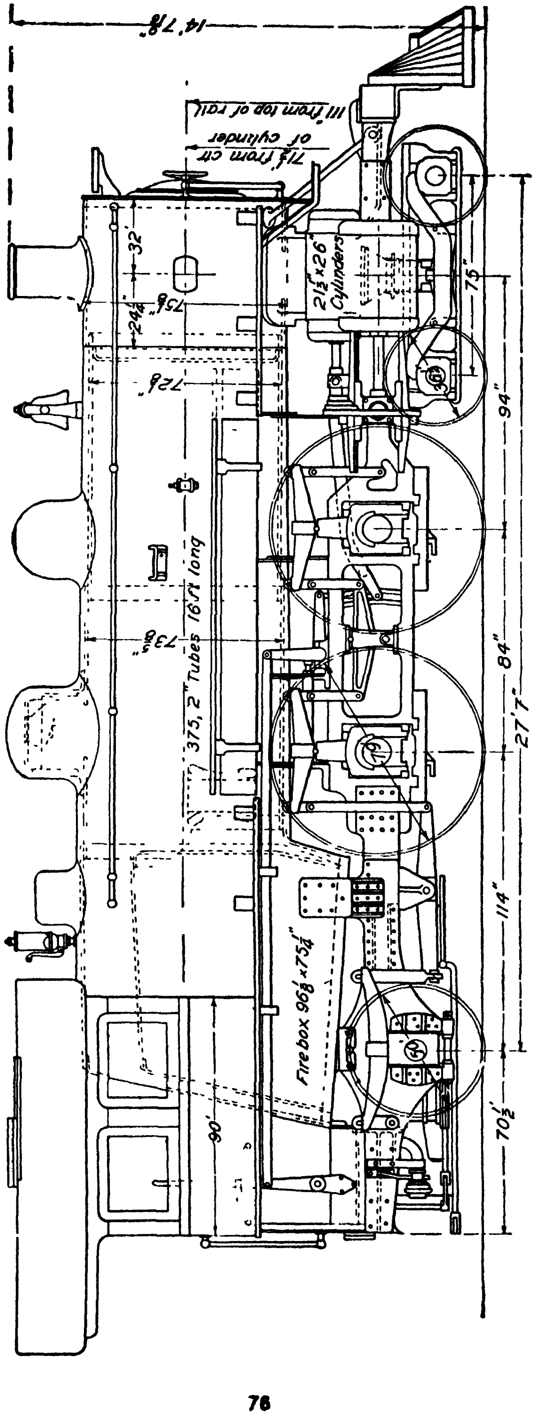


Fig 48 Detailed Drawing of Modern Locomotive Engine of the "Atlantic" Type

capacity. This engine is a compound type with piston valves, is well designed, is neatly proportioned, and admirably fulfills every requirement.

Compound Type. In connection with the subject of compounding just mentioned it may be said that in recent years the compound locomotive has been found in increased numbers on American railroads. A type of compound that has given especial good service and which is being adopted by many roads for heavy hill climbing duty is the Mallett Articulated Compound. The adoption of the compound locomotive has been due to a general opinion among railroad officials that the findings of a committee of the American Master Mechanics Association were true, as demonstrated by practice. This Committee says of compounding:

(a) It has achieved a saving in the fuel burned, averaging 18 per cent at reasonable boiler pressures

(b) It has lessened the amount of water to be handled

(c) The tender can, therefore, be reduced in size and weight

(d) It has increased the possibilities of speed beyond sixty miles per hour, without unduly straining the engine

(e) It has increased the haulage power at full speed

(f) In some classes of engines it has increased the starting power.

(g) It has lessened the valve friction per horsepower developed.

A number of other reasons are given in their report. Notwithstanding these facts, however, the compound locomotive has not come into very general use on railroads.

WATER PUMPS

The subject of pumping engines is a very broad one, and one which has received the thought and study of the most eminent engineers for many decades. From the earliest history of man there is gleaned the fact that human ingenuity and skill had been devoted in those early times to the perfection of some kind of power pump. It would be a difficult matter to mention an industry of any character or description but what a pump was needed somewhere in the enterprise. It was first used in a large way in the mining industries for pumping water out of the mines. Today it is found in all power houses, mines, and factories of various kinds. Both the large and small cities depend upon it for their water supply. The heating and ventilating systems of modern apartment houses and office

buildings use the pump, and mention might be made of many other instances where the water pump is indispensable.

There are two general classes of pumps, namely, crank or flywheel type and direct acting pumps.

Crank or Flywheel Type. The crank or flywheel type was the first form to be developed. These pumps vary greatly both in their design and in the details of their construction. They are of varying sizes, including some of the largest and most expensive in the world. As a general thing they are used in heavy hydraulic enterprises, for furnishing water supply for cities, and in various other enterprises where a large and constant supply of water is demanded. In this class of pumps or engines the application of the power in the steam cylinders in driving the pump plunger or piston varies greatly both in design and detail of construction. Long or short beams or bell cranks may be used and sometimes gearing may be employed, but in all cases the limit of the stroke of the steam piston and of the pump plunger is governed by the crank of a revolving shaft. In pumping engines it is not absolutely necessary to have a revolving shaft, the only requirement being that the piston in the pump cylinder shall be driven back and forth with a plain reciprocating motion which may be exactly like that of the steam piston. For this reason, in early pumping engines and also in modern engines, the reciprocating motion of the steam piston is applied directly, or through a beam, to produce the reciprocating motion of the pump piston or plunger without the use of any revolving part. Frequently, however, it is desirable to use a flywheel so that the steam may be used expansively, and in these cases, of course, a revolving shaft must be used.

Cameron Belt-Driven Pump. The power pump used as an illustration, Fig. 49, is a belt-driven one. The belt is placed on the pulley *1* and can be shifted to a loose pulley by the shifter *2*, when desired. The shaft *4*, which is driven by the belt pulley, extends across the frame and has attached to it a flywheel *5* and a small gear wheel, which meshes with the large gear wheel *3*. The gear wheel *3* is keyed to the crank shaft *6*, hence, when it is driven, the crank shaft is made to revolve, which in turn gives a back and forth movement to the piston as in the ordinary steam engine. The flywheel *5*, attached to the revolving shaft, may be of greater or less diameter and weight, depending on the condition under which the pump is to be operated.

In addition to assisting the crank to pass the dead center at each end of the stroke of the piston, it can be employed as a reservoir in which any excess energy may be stored at the beginning of each stroke and drawn out during the latter part of the stroke, where the force of the water column is greater than that of the steam. By this means it is possible to use shorter cut-offs in the cylinder than could otherwise be permitted; hence, a resulting saving in steam. Many means may be used to drive the power pump. While the illustration shows

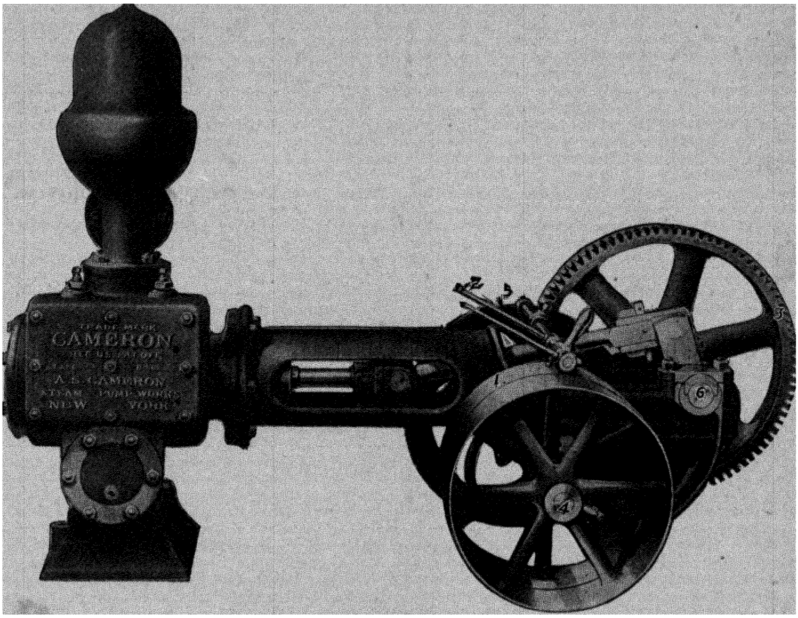


Fig 49 Belt-Driven Power Pump

one belt driven, yet they are frequently electrically driven, and sometimes the revolving shaft is attached to the shaft of a gas or steam engine.

Deep-Well or Mine Pump. For deep-well or mine pumping, the cylinders are often set in a vertical position directly over the pump cylinder. The piston rod extends from the steam cylinder directly below to the pump plunger. Sometimes it is possible to use steam expansively in these pumps by reason of the weight of the reciprocating parts. When the weight is sufficient, the steam can be cut off before the end of the stroke and the momentum of the parts will

be enough to just finish the stroke, consequently these pumps are sometimes compounded. They are used only in pumping from very deep wells.

Direct-Acting Type. A direct-acting steam pump is one in which there are no revolving parts, such as shafts, cranks, and fly-wheels, the power of the steam in the steam cylinder being transferred to the piston or plunger in the pump in a direct line through the use of a continuous rod or connection. In pumps of this construction there are no weights in the moving parts, other than those required to produce sufficient strength in such parts for the work they are required to perform and, as there is consequently no opportunity to store up power in one part of the stroke to be given out at another, it is impossible to cut off the steam in the steam cylinder during any

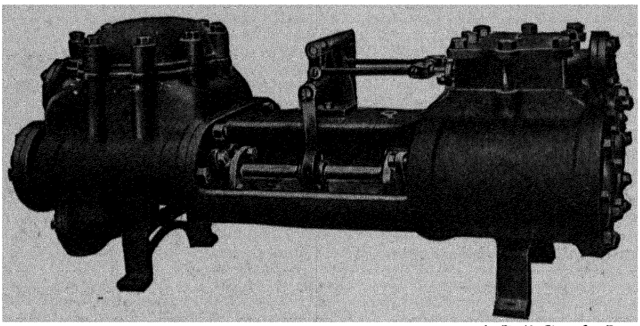


Fig 50 Direct-Acting Duplex Pump with Rocker and Bell-Crank Lever

part of its stroke. The uniform and steady action of the direct-acting steam pump is dependent alone on the use of a steady uniform pressure of steam through the entire stroke of the piston, against a steady, uniform resistance of water pressure in the pump; the difference between the power exerted in the steam cylinders and the resistance in the pump governs the rate of speed at which the piston or plunger of the pump will move. The length of the stroke of the steam piston within the steam cylinders of this class of pumps is limited, and is controlled alone by the admission, compression, and release of the steam used in the cylinders.

Duplex Pump with Rocker and Bell-Crank Lever. The direct-acting steam pump, Fig. 50, is known as a duplex pump and consists simply of two direct-acting steam pumps placed side by side. The steam pistons are at one end and the water pistons at the other.

The steam pressure acts directly on the pistons; no flywheel is used; and since the reciprocating parts are comparatively light and there is no revolving mass to carry by the dead points, it is evident that in the ordinary form there can be no expansion of steam. The pump is inexpensive and g ves a positive action. It uses a relatively large quantity of steam, but for small work the absolute amount is not very great.

On the piston rod of each pump is a bell-crank lever which operates the valve of the other pump. There must be a rocker on one side and a bell-crank lever on the other, because of the relative motion of the valves and pistons. The first piston, as it goes forward, must use a rocker, because it draws the second valve back. The second piston, as it goes back, must use a bell-crank lever because it must push the first valve back in the same direction as its own motion. The two pistons are made to work a half-stroke apart, thus one begins its stroke when the other is in the middle. In this way a steady flow of water is obtained, as both pumps discharge into the same delivery pipe. In large pumps of this kind, and even in some small ones, the motion described above merely admits steam to a small auxiliary piston on each steam cylinder, which then moves the main steam valve by steam pressure.

Duplex Pump with Tappet. Some pumps operate the steam valve by means of a tappet instead of a rocker and a bell-crank lever, Fig. 51. Its construction and operation is as follows:

A is the steam cylinder; *C*, the piston; *L*, the steam chest; *F*, the chest plunger, the right-hand end of which is shown in section; *G*, the slide valve; *H*, a lever, by means of which the steam-chest plunger *F* may be reversed by hand when expedient; *II* are reversing valves; *KK* are the reversing valve chamber bonnets; and *EE* are exhaust ports leading from the ends of the steam chest direct to the main exhaust and closed by the reversing valves *II*.

The piston *C* is driven by steam admitted under the slide valve *G*, which, as it is shifted backward and forward, alternately connects opposite ends of the cylinder *A* with the live steam pipe and exhaust. This slide valve *G* is shifted by the auxiliary plunger *F*, the latter having hollow ends which are filled with steam, and this, issuing through a hole in each end, fills the spaces between it and the heads of the steam chest in which it works. Pressure being equal at each

end, this plunger *F*, under ordinary conditions, is balanced and motionless; but when the main piston *C* has traveled far enough to strike and open the reverse valve *I*, the steam exhausts through the port *E* from behind that end of the plunger *F*, which immediately shifts accordingly and carries with it the slide valve *G*, thus reversing the pump. No matter how fast the piston may be traveling, it

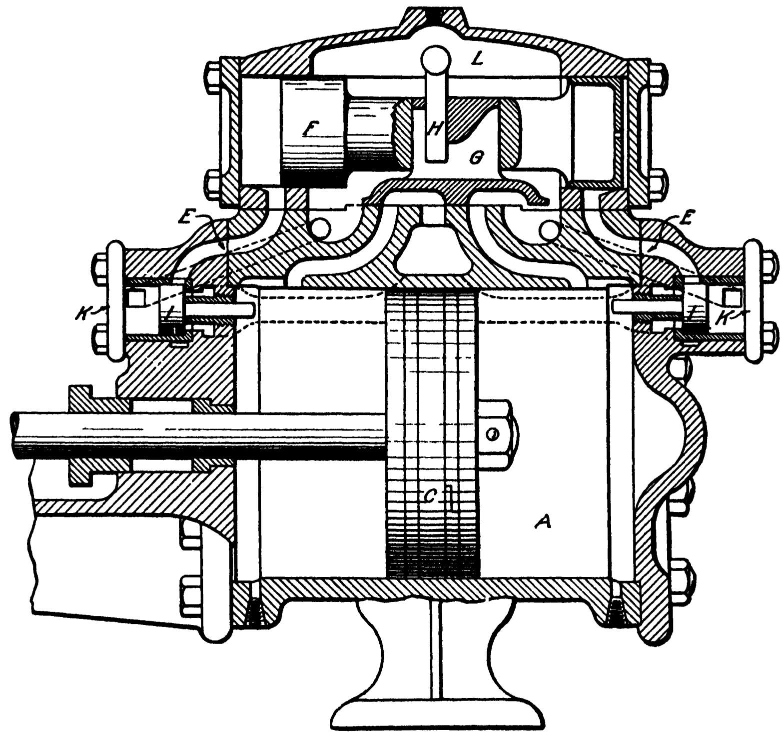


Fig 51 Section of Pump Cylinder Showing Valve Operated with Tappet

must instantly reverse on touching the valve *I*. In its movement the plunger *F* acts as a slide valve to close the port *E* and is cushioned on the confined steam between the ports and steam-chest cover. The reverse valves *II* are closed as soon as the piston *C* leaves them by a constant pressure of steam behind them conveyed direct from the steam chest through the ports shown by dotted lines.

The motion of the piston *C*, Fig. 52, is transmitted through the rod *M* to the water piston in the cylinder *R*. As the piston moves

back and forth, water enters through the intake valves *O* and leaves through the discharge valves immediately above, and finally leaves through the delivery pipe *P*. In order to create a more continuous flow of water, an air chamber *Q* is provided. Any sudden variation in the pressure in the line is taken up largely by the air chamber. It also serves to lessen the effect of water hammer.

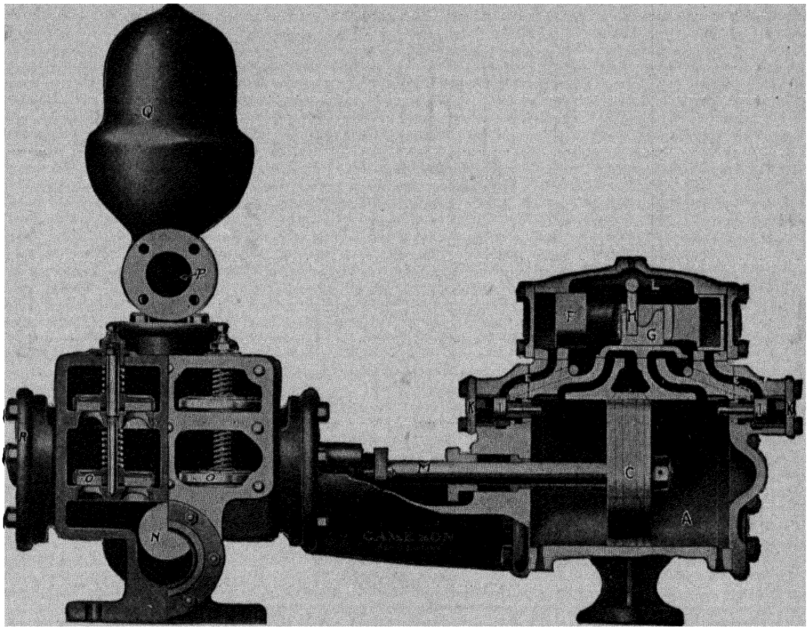


Fig 52. Section of Duplex Pump with Tappet

SPECIAL ENGINES

Under this heading may be placed a large number of engines which have been built for a very definite field of usefulness, such as various types of fire engines and automobile engines, where steam is used as the motive force. Again a number of experimental engines have been built, commonly known as freak engines, having peculiar construction and design, which never got beyond the experimental stage. Rotary engines as well as rotary pumps have been used to some extent, but the rotary engines thus far developed have been so extravagant in steam consumption that their use has been discontinued. It is thus seen that under the head of special engines many

of the engines already discussed, as well as an untold number of others of more or less merit, may be properly classed.

The special engines referred to above were not mentioned for the purpose of studying them, but rather to indicate that outside and distinct from the steam engines classified and considered, there are a large number of special types that should not be entirely ignored.

MARINE ENGINES

The subject of Marine Steam Engines is a broad and important one, and to treat it properly would require one or two volumes the size of this one. However, it seems desirable to discuss the subject in a very general way in connection with the still broader subject of The Steam Engine, and thus give the student a general idea of marine engine parts.

Definition of Terms. Before taking up the subject, it is thought advisable to present a brief statement of nautical terms used in

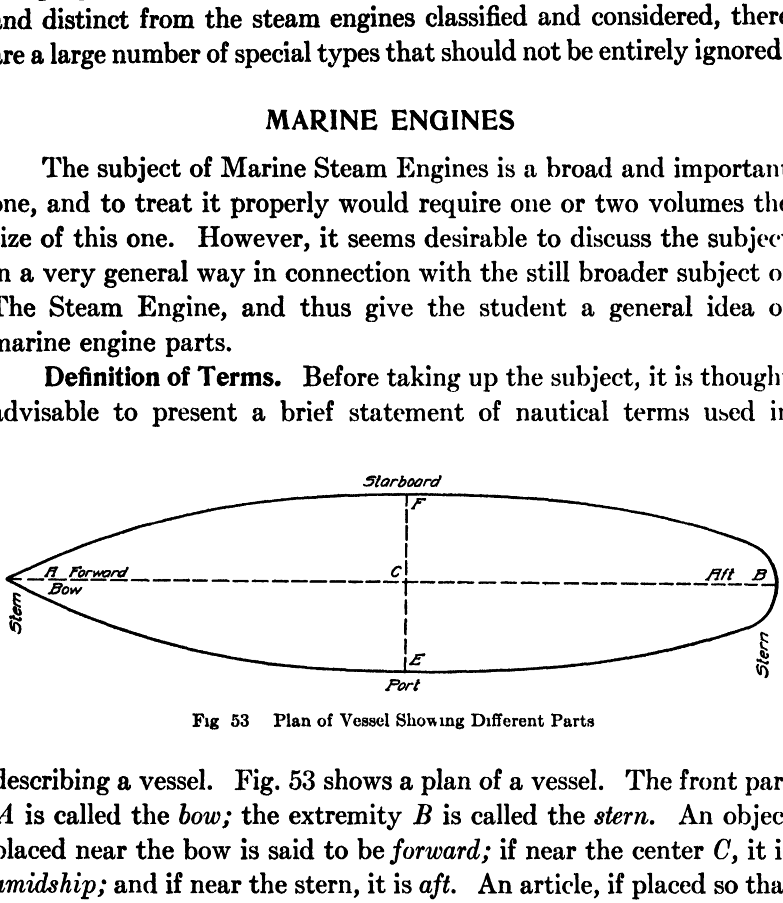


Fig 53 Plan of Vessel Showing Different Parts

describing a vessel. Fig. 53 shows a plan of a vessel. The front part A is called the *bow;* the extremity B is called the *stern.* An object placed near the bow is said to be *forward;* if near the center C, it is *amidship;* and if near the stern, it is *aft.* An article, if placed so that its major dimension is parallel to the line AB, is said to be placed *fore and aft.* Thus the crankshaft of a triple expansion engine of a vessel is located along the line AB and is sometimes spoken of as a fore-and-aft engine. An article located crosswise of the vessel, that is, at right angles to AB, is said to be placed *athwartship.* To one standing on the deck facing the bow, the *starboard* side is on his right and the *larboard,* or *port* side, on his left.

The width of a vessel FE is its *beam,* and the perpendicular

distance from its lowest part to the surface of the water is called the *draft*. The length of a vessel is the horizontal distance between perpendiculars drawn at its extreme ends. The displacement of a vessel is equal to the weight of water it displaces and is usually expressed in long tons.

The *speed* of a vessel is usually expressed in *knots* per hour, but is sometimes given in miles per hour. A knot is equal to about $1\frac{1}{6}$ miles.

Methods of Propulsion. Speaking in a general way, the propulsion of a steam vessel is accomplished by causing a mass of water adjacent to the ship to move in a direction opposite to that of the ship. Motion is imparted to the water in one of the following three ways: (1) by paddle wheels; (2) by screw propellers; and (3) by jets of water or hydraulic propulsion.

The oldest of these three forms of propulsion, the paddle wheel, is still much used in lake and river steamers and ferry boats; but for ocean-going vessels and in many boats on inland waterways, the screw propeller has supplanted it. Jet or hydraulic propulsion has not proved to be practical and for this reason has never been used in commercial work.

TYPES OF ENGINES

Beam Type. The first steam vessels were fitted with paddle wheels, and as beam engines were the most common, this form of engine was used. Its construction, however, was somewhat modified for this service. This arrangement of beam engine and paddle wheel was used for many years and was applied to ocean vessels as well as to small river boats. It is still used, especially in this country, on river steamers and some coast steamers. The beam is supported by a large A-frame on the deck, and the engines are about on a level with the shaft.

Engines of this type take up rather more room than those now in common use, partly because of great size, and also because of the shaft and paddle wheels. Another disadvantage is that in heavy weather when one paddle wheel is thrown out of the water the other is deeply immersed and takes all the strain, so that there is a tendency to rack the boat. Then again if the boat is loaded heavily, the paddle blades are very deeply immersed; while if light, they barely

touch the water. It is difficult to handle the engines satisfactorily under either condition.

Inclined Type. The introduction of the screw propeller overcame these difficulties very largely and at the same time required a high speed engine. At first, the increased speed was supplied by the use of spur-wheel gearing, but gradually higher speed engines were built and connected directly to the propeller shaft. It was, of course, difficult with small width at each side of the shaft to use horizontal engines, therefore various arrangements of inclined engines were used before the vertical engine was finally chosen by all as the standard form for marine work. It is only in recent years that the vertical engine has become general in naval work and in merchant steamers.

Vertical Type. In merchant ocean steamers the common form has three cylinders set in line, fore and aft, above the shaft, the cranks being set 120 degrees apart in order to give a more even turning moment. The three cylinders are worked triple expansion, the valves being usually of the piston type on the high and intermediate and double-ported slide type on the low. Sometimes piston valves are used on all the cylinders. Plain slide valves are not suitable for high-pressure work of any kind. While steam turbines are used to some extent in ocean-going vessels, the majority of ships in this service are equipped with high-speed, vertical, multicylinder engines direct connected to the propeller shaft.

Cylinder Arrangement. The different arrangement of marine engine cylinders commonly found in service is shown in Figs. 54 to 57.

Tandem- and Cross-Compound Types. In Fig. 54, *A* is the tandem-compound arrangement with its single crank; *B* is the cross-compound with cranks set 90° apart; and *C* is the three-cylinder compound with cranks set 120° apart. In arrangement *C*, the high-pressure cylinder is sometimes placed between the two low-pressure cylinders.

Triple-Expansion Type. The cylinder arrangement, Fig. 55, is found only on the larger vessels, and is spoken of as the triple-expansion type. In this type there are three cylinders to each engine, and they are called the high-, intermediate-, and low-pressure cylinders, each succeeding one being of larger volume than the one preceding. Fig. 55 illustrates two arrangements of the cylinders of triple-expansion engines. In arrangement *A* the cylinders follow

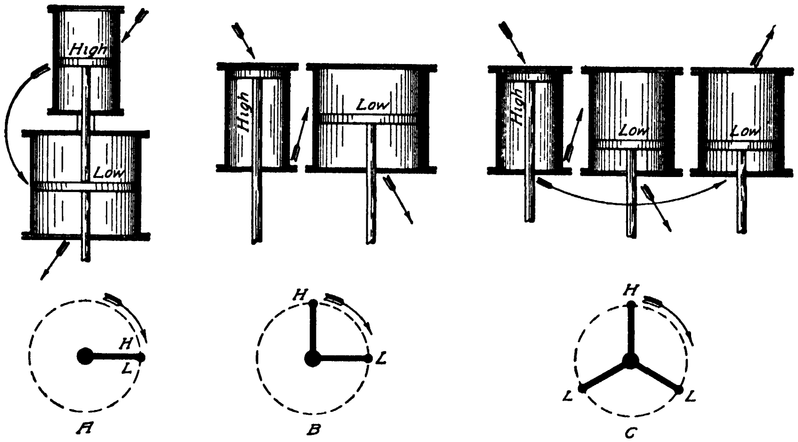


Fig 54 Diagrams of Tandem and Cross-Compound Cylinder Arrangements

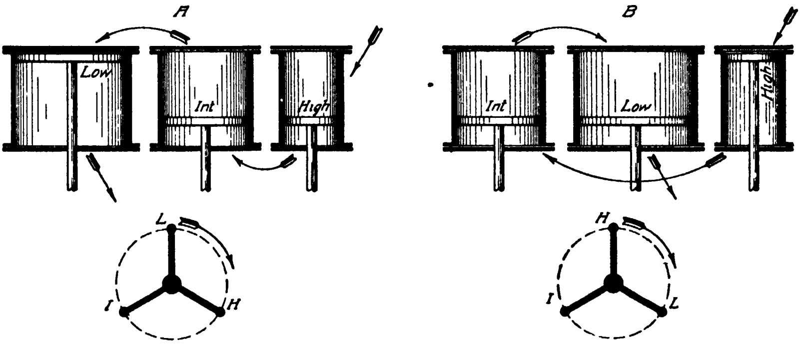


Fig 55 Diagrams Showing Triple-Expansion Cylinder Arrangements

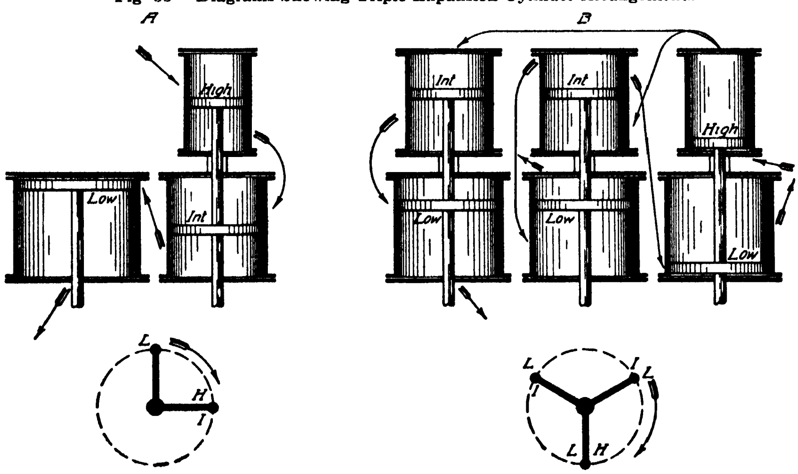


Fig. 56 Diagrams Showing Other Triple-Expansion Cylinder Arrangements

each other in natural sequence; this requires the least length of piping. Arrangement *B* is frequently used, but requires more piping than arrangement *A*. Another common arrangement is to put the high-pressure cylinder in the center of the group. In any of these systems the cranks would be set at 120°, giving a more nearly uniform turning movement to the shaft, since each cylinder will develop approximately one-third the total horsepower of the engine.

Still other arrangements of the cylinders of triple-expansion engines are found in Fig. 56. Arrangement *A* gives the effect of a tandem-compound between the high- and the intermediate-pressure cylinder and a cross-compound between these two and the low-pressure cylinder—an arrangement which results in cranks being

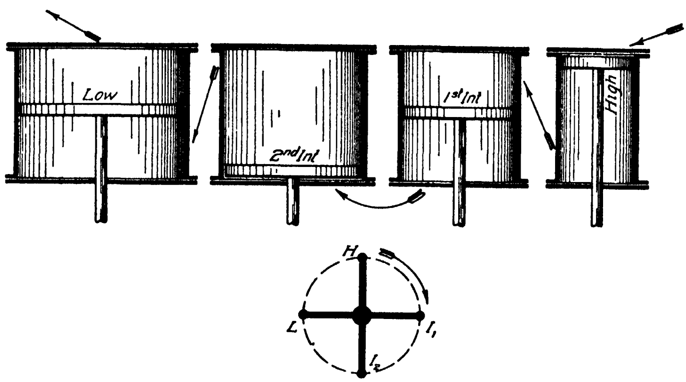


Fig 57 Diagram Showing Quadruple-Expansion Cylinder Arrangement

set at 90° with the consequent uneven turning effect, but it is sometimes resorted to because of lack of space for all three cylinders in line. Arrangement *B* is a triple-expansion engine, having six cylinders. Here the volume of one intermediate-pressure cylinder is divided among two cylinders, and the volume of one low-pressure cylinder among three cylinders. This form is very expensive and is not often used. The arrangement requires less floor area than would be required for the same power in a three-cylinder engine.

Quadruple-Expansion Type. The last cylinder arrangement to be considered is found on the quadruple-expansion engine. In this type the steam goes from the high-pressure cylinder to the first intermediate, then to the second intermediate, and finally to the low-pressure cylinder. The volume of each cylinder is larger than that of

the preceding cylinder. There are many different arrangements of cylinders possible with quadruple-expansion engines. Fig. 57 shows the arrangement of cylinders in their natural sequence with the four cranks set 90° apart, which gives a slightly more even turning effort

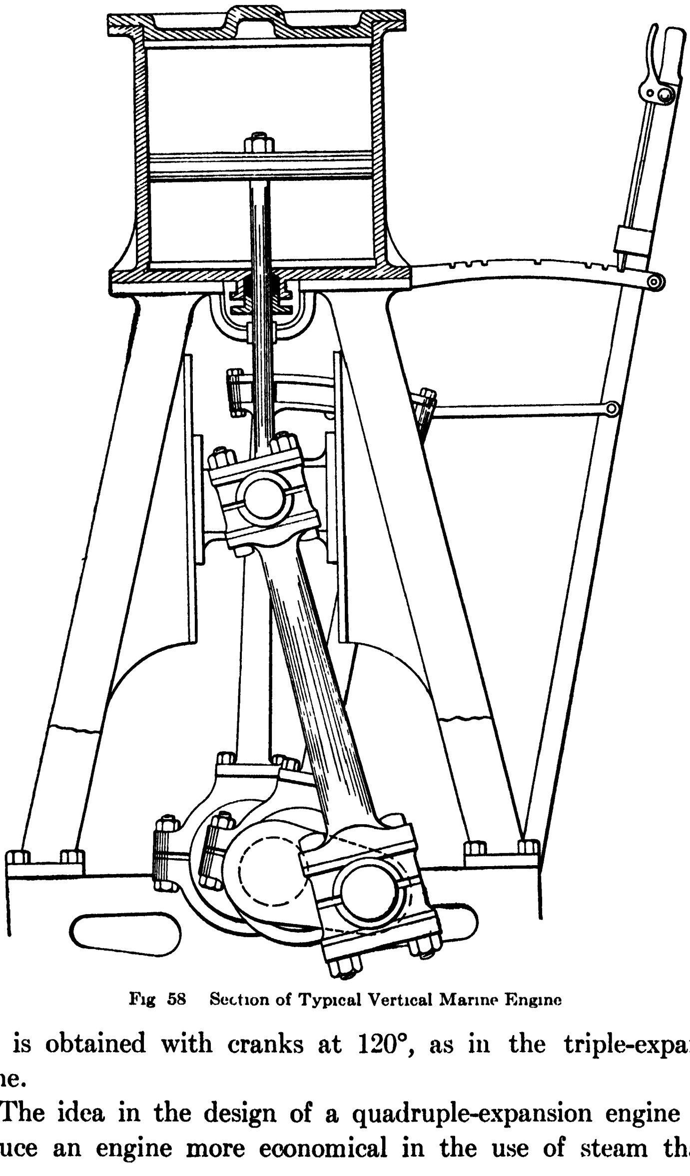


Fig 58 Section of Typical Vertical Marine Engine

than is obtained with cranks at 120°, as in the triple-expansion engine.

The idea in the design of a quadruple-expansion engine is to produce an engine more economical in the use of steam than is

obtained in any other type. With high-pressure steam, say 200 pounds and over, it gives a better economy in the use of steam than does the triple-expansion engine. However, the saving effected in the use of less steam is, to a very large extent, offset by an increase in first cost, operating cost, and general upkeep.

Comparison of Marine with Stationary Types. Fig. 22, page 29, and Fig. 58 show cross-sectional views of marine engines. In marine work many different designs of engines are used. These two views are intended to present merely the general features and characteristics of the marine engine. In comparison with stationary engines attention is called to the different form of frame used, lighter frames, different details of the connecting rod, and in the latter figure the separate crankshaft for each cylinder and the single crosshead guide. Also the cylinders are of complicated form and have double walls, and the pistons are of a cup shape. These points will be brought out more in detail in what follows.

ENGINE DETAILS

Cylinder. The general type of steam cylinder for a marine engine consists of three distinct parts, namely, the shell, the liner, and the cover.

Shell. In Fig. 59, the shell is the outer casting forming the outside cylinder wall, the lower cylinder head, and the steam ports. As its complicated form makes the casting of the shell a difficult matter, an iron is used that runs freely in the mould. Sometimes the lower cylinder head is not cast integral with the shell, but is fitted to it separately like the cover.

Liner. The liner is the plain cylindrical casting or bushing, which forms the inner cylinder wall. Its use is made necessary because the metal in the shell is of such composition that it will not wear well if the piston is permitted to work directly on it. The material of the liner is usually hard, close-grained cast iron. In some cases forged steel is used. It is secured to the shell by bolts through a flanged end, or by stud bolts. But one end is fastened to the shell, the other end being left free to expand under the influence of the higher temperatures to which the liner is exposed.

Cover. The cover forms the upper end of the cylinder. Usually it is made of steel to combine lightness and strength. Sometimes the

cover is cast hollow so as to form a steam jacket for the cylinder head, but more often it is made of a single wall of metal, reinforced by radial ribs on the outside.

Marine Details Resemble Stationary. Many of the details of marine engines are so nearly like those of stationary engines in essential features, and the minor points of difference are so varied that special mention of them will not be made here. For illustrations of different parts, reference may be made to the earlier sections of this book. For example, Fig. 9 shows a typical marine piston and connection to the piston rod; Fig. 14 a typical crosshead and crosshead pin; Fig. 16 a connecting rod; and Figs. 18 and 19 typical main bearings.

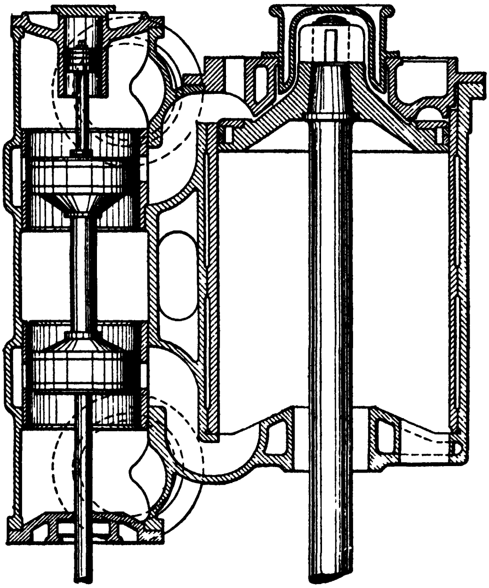


Fig 59 Sectional View of Marine Engine Cylinder, Piston and Steam Ports

Crosshead Guides. A form of marine crosshead guide, differing from that ordinarily used in stationary work, is shown in Fig. 60. The crosshead used with this guide is known as the slipper type. It has but one bearing surface, and this runs in the space between *SS* and *A*. In the guide the plate *P* is bolted to the engine frame so that it receives all the crosshead pressure when the engine is running ahead. For backward motion of the engine the flanges *FF* are provided to receive the thrust.

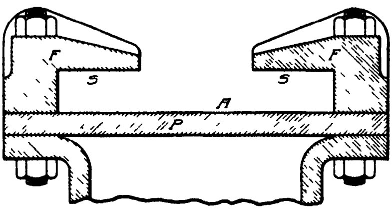


Fig 60 Type of Marine Crosshead Guide

Cranks. In marine work side cranks are not used. The connecting rod is always connected between two crank arms. Further-

more, each cylinder of an engine has a separate crankshaft. These separate shafts are bolted together by flanges, as shown in Fig. 61. The dotted lines in this figure show how, in large shafts, the center is sometimes made hollow. This is done to make a saving in weight and to remove inperfect portions usually found in the center. The

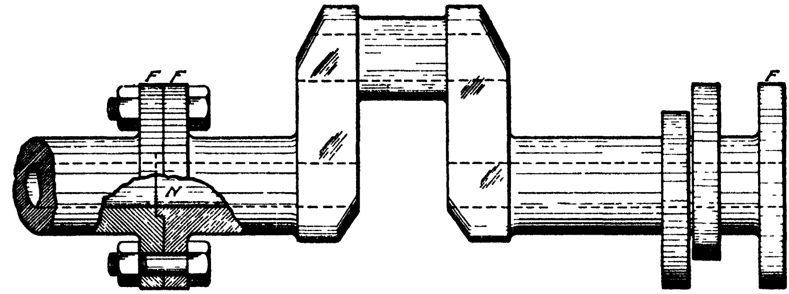


Fig 61 Portion of Marine Engine Crank Showing Method of Bolting Sections Together

center of the shaft is the least effective of any part of it in resisting twisting forces, and the outside is the most effective. By using a little larger shaft, therefore, and removing considerable metal from around its center, a shaft of the same strength as a solid one is obtained, with a material saving in weight. The crankshafts are

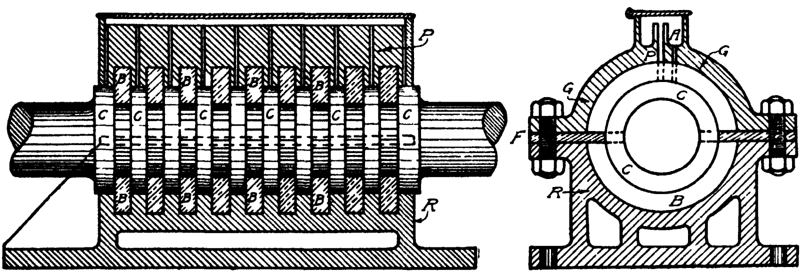


Fig 62 Typical Marine Thrust Bearing

usually all made of the same size, so as to be interchangeable, and thus require fewer parts to be kept in stock.

Bearings. The bearing, Fig. 62, while not part of the engine, will nevertheless be discussed at this point. This is called the thrust bearing, and is used to relieve the engine of the thrust caused by the revolving propeller in screw-propelled vessels. The propeller shaft

is turned with the collars *C* as a part of it. The cast-iron box *R* is secured to the frame of the vessel just aft of the main engines, and the cap *G* is bolted to it, as shown. The collars *C* press against rings *B* of gun metal or brass and transmit the propeller thrust to them, and thence to the vessel. Rings *B* are split and are prevented from turning by the tongue piece *F*. Holes *P* are for lubrication and holes *A*

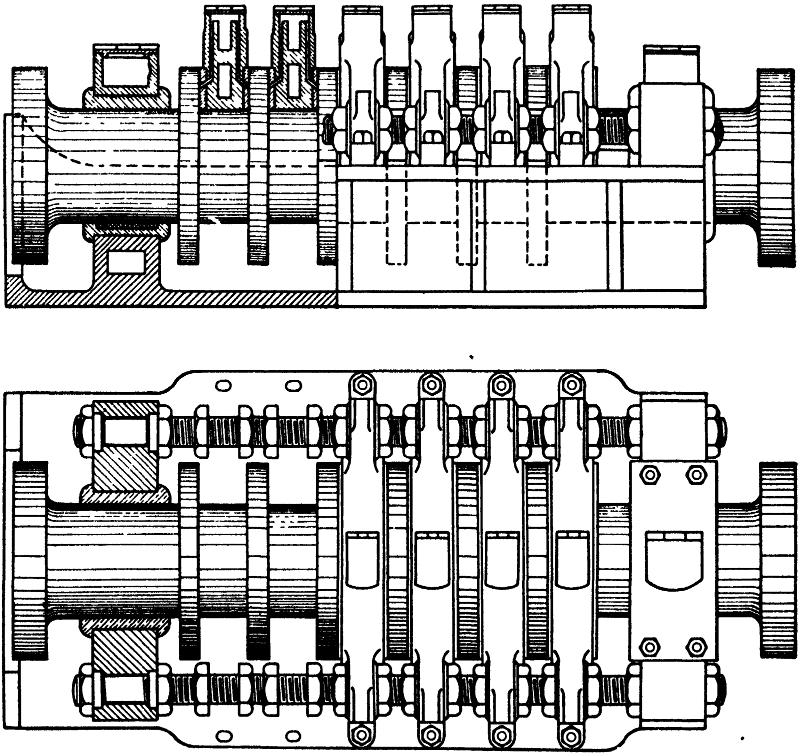


Fig 63 Type of Thrust Bearing in Which Provision Is Made for Taking Up Wear

are provided for water cooling when needed. Water may also be circulated through the base.

Fig. 62 shows the principle of the thrust bearing, but it is not much used because no provision is made for taking up unequal wear between the brasses. Fig. 63 shows a type of bearing in which provision is made for this feature, the wear being taken up by means of the nuts fitted to the long screws at either side of the thrust bearing.

Thrust Bearing Calculations. The number of collars required in any given thrust bearing depends primarily on the total thrust that will come on them. There may be a large number of collars of small diameter or a small number of large diameter. The experience of the designer is usually the determining factor as to the number used.

Knowing the number of collars required, their diameter may be computed from the following formulas, in which n is number of collars; D is diameter of collars; d is diameter of shaft; P is total thrust; and p is safe allowable pressure per square inch of area, which is usually taken as 60 pounds per square inch.

First taking the formula expressing the total thrust, we have

$$P=p\left(\frac{\pi D^2}{4}-\frac{\pi d^2}{4}\right)n$$

and substituting for p the value of 60 pounds per sq. in., there results the formula

$$P=60\times\frac{\pi}{4}(D^2-d^2)\,n$$
$$=47\,(D^2-d^2)\,n$$

Transposing in the last formula and solving for the value of D, there results the equation

$$D=\sqrt{d^2+\frac{P}{47n}}$$

which gives the diameter of collars required for the conditions assumed.

AUXILIARY APPARATUS

The auxiliary apparatus aboard a ship is far more numerous than would be suspected by one not acquainted with it or even by one familiar with the apparatus in stationary power plants. The general features of some of the more important pieces of apparatus, only, will be described.

Reversing Mechanism. The reversing mechanism of large marine engines is so large and heavy and, at times, has to be moved so quickly that it cannot be done by hand. Consequently, in some instances a small steam power cylinder is attached to the reversing gears to move them. This apparatus is called the steam-starting gear and is under the control of the engineer.

The action of this gear, Fig. 64, is as follows: When the reversing lever, or handle, is moved from the mid-position A to B, the rod CE is moved to the left. This movement raises the rod H, which is connected to the lever fulcrumed at T. As the rod H raises, the rod

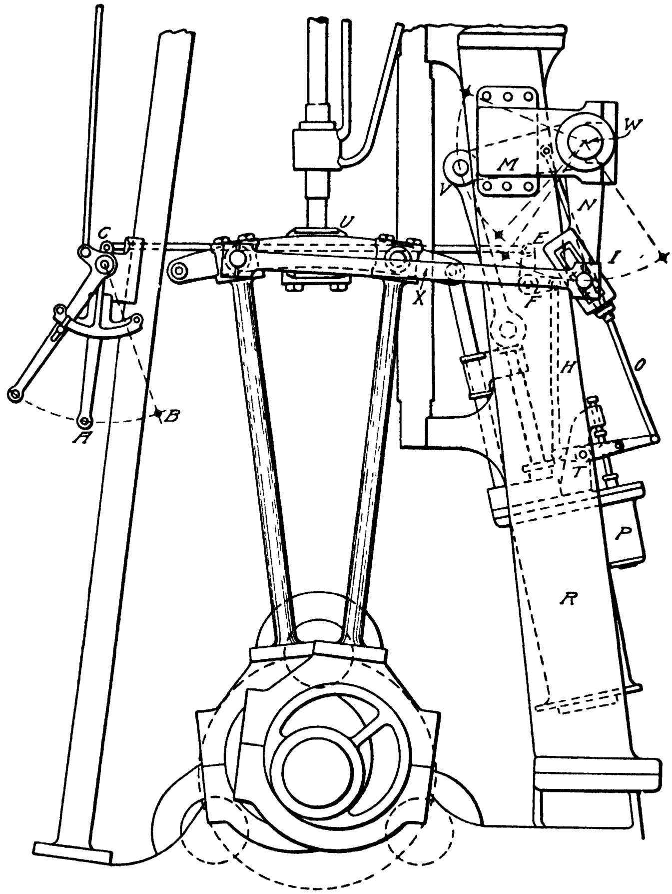


Fig. 64 Details or Steam Starting Gear

O moves downward, thus causing the arm M to move downward and the arm N to move to the right. This movement of the arm N and pin I causes a corresponding movement to the right of the reach rod and link, to which it is connected. Thus it is readily seen that the movement of the reversing lever A moves the link slightly and at the

same time causes steam to be admitted to the power cylinder, which acts on the piston and aids in the movement of the links.

Condensers. *Surface Type.* In marine work the surface condenser is used almost exclusively, because with this type the cooling water (sea water) does not come in contact with the steam, and the latter can then be used over and over in the boilers. Jet condensers on ocean vessels would prevent the continued use of the condensed steam because of the deposit the salt of the water would leave on the boiler tubes and shell.

Keel Type. In small boats, such as steam launches, the surface condenser would occupy much valuable room and add considerable weight, so a substitute, called the keel condenser, is frequently used. This consists of several rows of copper tubes placed outside the hull along the keel of the boat. The engine exhaust enters at one end of these tubes, is condensed by the sea water in contact with the outside of the tubes, and is then drawn out of the condenser by the air pump and pumped back to the boiler. This form of surface condenser requires no circulating pump.

Pumps. *Centrifugal Type.* The pump most often used on shipboard to circulate condenser cooling water is of the centrifugal type, driven by an independent engine or motor. The absence of valves in this kind of pump is of advantage, as is also the fact that it can be run through a greater range of speed and, consequently, give greater volumes of water when occasion demands. Oftentimes the piping is so arranged that these pumps can draw from the engine room bilge and discharge without passing the sludge through the condenser.

Air and Vacuum Types. Of the many kinds of air or vacuum pumps used, the one shown in Fig. 65 has been chosen for description as being a good example and one easy to understand. The operation of the pump is as follows: The inlet E is piped to the outlet of the condenser. On the up-stroke of the piston P, a partial vacuum is formed below it, enabling the condensed steam and air in E to rush through the foot valves F and into the pump cylinder B below the piston. After reaching the upper limit of its stroke, P descends, producing a slight pressure on the air and water entrained in the cylinder, which closes the foot valves against the escape of the cylinder contents. As the piston continues, the bucket valves H in the

piston are forced open, permitting the escape of air and water to the space above. On the next up-stroke this air and water are forced out of the air pump through the delivery valves *A* and the outlet *N*.

A small check valve, or pet-cock (not shown in the figure), is usually located in the cylinder wall *B* just below the delivery valves. When insufficient air comes through with the condensed steam to properly cushion the piston on the up-stroke, this valve is opened to provide the required air for cushioning. This air does not affect the degree of vacuum, because it is on the discharge side of the pump, where pressure on the piston is immaterial as regards vacuum in the condenser.

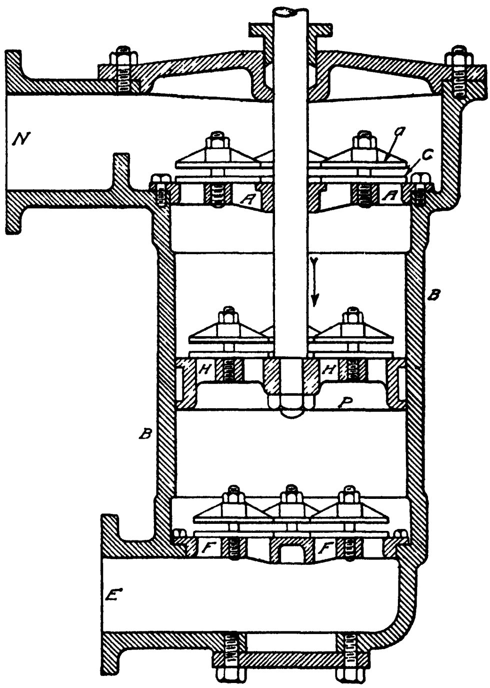


Fig 65 Section of Air or Vacuum Pump

Vacuum is measured by gages similar to those used for measuring high pressures, but calibrated to read in inches of mercury instead of pounds per square inch. A column of mercury under atmospheric pressure will stand about 30 inches high. Consequently, since 30 inches of mercury is equal to about 15 pounds per square inch, one inch of mercury will be equal to 15 divided by 30, or nearly one-half pound per square inch, the exact value being 0.49 pounds per square inch. Suppose the vacuum gage of a condenser reads 26. This means there has been a reduction of pressure corresponding to 26 inches of mercury or 26×0.49, or 12.74 pounds per square inch. If the atmospheric pressure is 14.7 pounds per square inch, then there remains in the condenser $14.7 - 12.74$, or 1.96 pounds per square inch absolute pressure.

Besides the auxiliary apparatus already mentioned, there are

many more on large and small vessels which cannot be discussed here, such as machines used for ventilation, forced draft, steering, weighing anchor, operating hoists and capstans, compressed air and refrigeration machines, and electric lighting.

PROPULSION

Process of Starting. When the engines are started and the screws or paddle wheels of a ship begin turning, there is no appreciable motion of the ship for a short time. During this short time the work done by the propellers is all used in overcoming the inertia of the vessel. As the inertia is overcome, the ship gradually begins to move and increase its speed. As the speed increases, the resistance offered to the motion of the ship through the water also increases. When all the power of the screws or paddle wheels is used in overcoming the resistance of the water to the passage of the ship through it, then the ship will be moving along at an approximately constant speed.

Resistance Factors for Ship in Motion. In smooth, quiet water the resistance offered to the ship's motion may be divided into three elements, namely: (1) frictional resistance of the hull; (2) eddy-making resistance; (3) wave-making resistance.

The most important of these is the frictional resistance, or skin friction. The amount of this resistance depends on the area and the length of the immersed surface of the hull, the roughness of this surface (whether covered with barnacles, sea-weed, etc.), and the speed of the ship.

Eddy-making resistance, which is usually small, is caused by eddy currents following just astern of the ship and by the churn of the propellers.

Wave-making resistance is caused by the waves made at the ship bow.

Winds and waves also offer resistance to a ship, but the amount of resistance due to these causes is difficult to estimate.

Variations of Resistance with Speed of Vessel. It has been shown by experiment that for a given ship, the resistances vary almost directly as the square of the speed, and that the power required to overcome these resistances varies almost as the cube of the speed. That is, if at a speed of 10 knots an hour a ship encounters a certain

resistance R and requires a certain power P, if the speed be increased to 20 knots, the resistance will be increased to R^2 and the power to P^3.

Indicated Thrust. Indicated thrust is a mathematical expression denoting the ratio of the total work in foot-pounds done by the main engines to the distance through which this force acts. Expressed as a formula, this ratio becomes

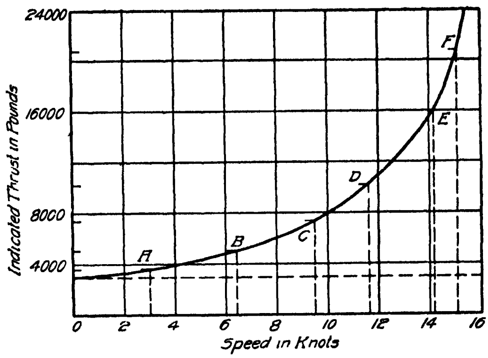


Fig 66 Curve Showing Indicated Thrusts for Different Speeds

$$T=\frac{33,000\times I.H.P.}{pN}$$

where T is indicated thrust in pounds; $I.H.P.$ is indicated horse-power of engines; p is pitch of screw in feet; and N is number of revolutions per minute. Since $I.H.P.=\frac{2\,PLAN}{33,000}$, this formula may be reduced to the form

$$T=\frac{2\,PLA}{p}$$

Where P is equivalent mean effective pressure in pounds per square inch; L is length of stroke in feet, and A is area of low-pressure piston in square inches.

EXAMPLE. What is the indicated thrust of a 1200 I H P marine engine driving a propeller of 20-foot pitch at 90 r p m ?

SOLUTION.

$$T=\frac{33,000\ \text{I H P}}{pN}$$
$$=\frac{33,000\times1200}{20\times90}$$
$$=22,000 \text{ pounds}$$

The indicated thrust for any given ship may be taken from a curve, such as is shown in Fig. 66.

Economical Speed. The most economical speed of a ship is that speed at which it can travel a given distance with the least consumption of fuel. At speeds either above or below this particular speed, the fuel consumption will be increased. To determine the most

economical speed, the amount of coal used at different speeds is determined by trial. These amounts are then plotted, as shown in Fig. 67. As it stands, this curve shows merely the coal consumed at different speeds, but by drawing a line from *O* tangent to the curve, the most economical speed is found at the point of tangency, or in this case at *N*, or about 8 knots per hour. If the coal used by the auxiliary machinery is to be considered, then *OX*, the amount of this coal, is laid off as shown, and the tangent drawn from the new

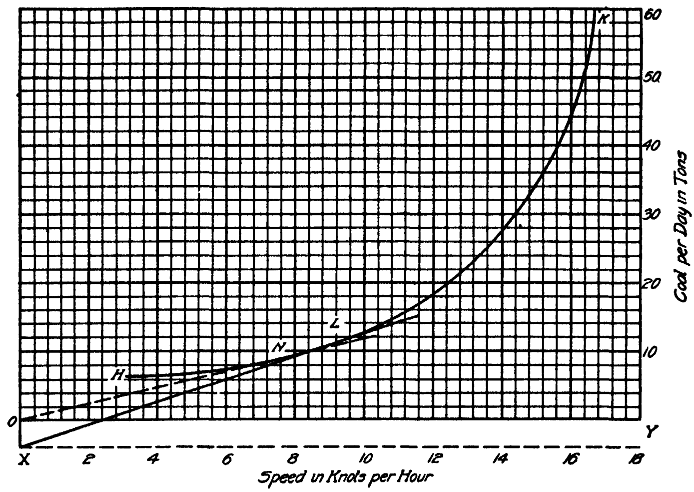


Fig 67 Curve Plotted to Show Most Economical Speed of a Ship

origin at *X*. This new line, tangent at *L*, gives a higher speed for the most economical one than that given for the main engines only.

PROPELLERS

Although propellers are not, strictly speaking, a part of marine engines, yet the two are so closely related that a brief discussion at this point seems desirable. Screw propellers only will be considered, because they are used more extensively than any other device for propelling vessels of various kinds.

Details of Screw Propeller. A screw propeller is a set of blades, usually constructed of iron or bronze, which are made to revolve in the water at the stern of the ship, by being connected to an extension of the main engine shaft.

Small propellers are usually cast with the hub and blades in one piece, but large ones have a central boss to which the blades are bolted. Propellers are made of a variety of metals, including iron, steel, bronze, and gun metal.

Blades. The general appearance of a blade may be seen from Fig. 68. Propellers may have two, three, or four blades. In merchant vessels the latter is most common.

Pitch. The pitch of a screw propeller is the distance in the direction of the axis of the screw that would be traveled by a point on the blade during one revolution if there were no slip. It is similar to the pitch of an ordinary lathe feed screw, but of course is much larger.

Diameter. The diameter of a screw propeller is simply the diameter of a circle described by the extreme ends of the blades. The ratio of the diameter to the pitch of a propeller is ordinarily from 1 to 1.1 and up to 1 to 1.5. Thus for a 14-foot diameter propeller the pitch would likely be from $14\times1.1=15$ feet to $14\times1.5=21$ feet.

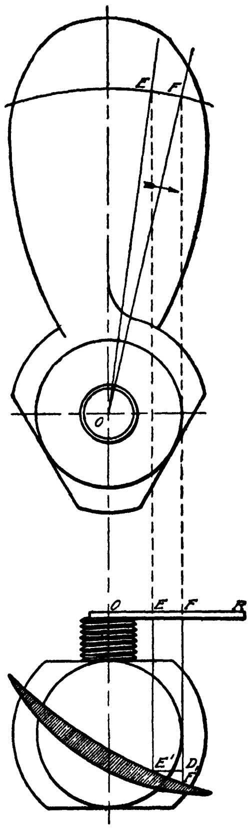


Fig 68 Typical Shape of Propeller Blade

Propelling Action of Screw Propeller. When a screw propeller is revolving in a given direction (for go-ahead motion for instance), the blades press on the water as the threads of an ordinary screw do upon the threads in the nut. The pressing of the blades on the water causes the water to be driven backward. There is, however, a reaction caused by projecting this mass of water sternward which results in the ahead motion of the boat. The useful work done by the propeller is the work which forces the water directly sternward; of course, the movement of water in any other direction than sternward results in a waste power.

If the screw worked in an unyielding medium, it would advance a distance equal to its pitch at each revolution. Hence, the speed of

the screw per minute is the product of the pitch and the number of revolutions per minute.

EXAMPLE Suppose a screw is of 18-foot pitch and makes 72 revolutions per minute What is the speed of the screw in feet per minute and knots per hour?

SOLUTION.

$$18 \times 72 = 1{,}296 \text{ feet per minute}$$
$$1{,}296 \times 60 = 77{,}760 \text{ feet per hour}$$
$$\frac{77{,}760}{6{,}080} = 12\ 78 \text{ knots per hour}$$

Slip. Water is a yielding medium and for this reason the pressure of the blades causes the water acted on to be driven back instead of remaining firm. Then the actual speed of the ship (when referred to the undisturbed water at a slight distance from the ship) is less than the speed of the screw. This difference is called slip. *Slip is the difference between the speed of the screw and the speed of the ship, relative to still water.* It is expressed in feet per minute and as a per cent of the speed of the screw.

EXAMPLE A ship is moving at the rate of 16 knots per hour The screw has a pitch of 19 feet and makes 97 revolutions per minute What is the slip?

SOLUTION.

$$19 \times 97 = 1{,}843 \text{ feet per minute} = \text{speed of screw}$$
$$\frac{16 \times 6{,}080}{60} = 1{,}621 \text{ feet per minute} = \text{speed of ship}$$
$$\text{Slip} = 1{,}843 - 1{,}621 = 222 \text{ feet per minute}$$
$$= \frac{222}{1{,}843} = 1204 = 12\ 04 \text{ per cent}$$

This may be expressed algebraically as follows: Let S equal speed of screw; s equal speed of ship; and L equal slip in feet per minute. Then

$$L = S - s$$

$$\frac{S-s}{S} \times 100 = \text{slip expressed in per cent}$$

The slip thus found is not the actual slip, but the apparent slip. It is not the actual or real slip, because the screw does not act in still

water, but in water that has been set in motion by the screw itself or by the hull.

While the hull moves through the water, it sets in motion the water in contact with it, the direction being the same as that of the ship. The water close to the ship has a greater forward velocity than that at a distance. Since this water has a velocity a little less than that of the ship, it soon falls behind the hull and is found at the stern. Thus the water in which the propeller acts has a forward velocity. Also the velocity is influenced by the waves and eddies, due to the lines of the vessel. On account of the many conditions that make the velocity of the wake variable, it is difficult to calculate it.

When the propeller is considered, it is evident that the condition of the water in which it works should be considered. Since the velocity is difficult to obtain, the real slip is not easily found.

When slip is referred to, it is generally the apparent slip that is intended and not the real slip. The apparent slip varies from 5 to 25 per cent—15 to 20 per cent being a fair average. The actual slip is usually from 5 to 15 per cent greater than the apparent.

MANAGEMENT OF MARINE ENGINES

It is of great importance that the chief engineer and all of the assistants should be familiar with the machinery of the ship. The steam and exhaust pipes, both main and auxiliary, and the location of the valves should be carefully traced; also the feed pipes to the boilers, and the piping to the condensers. It is important that each officer should know the function of every pump and the piping from the bilges. Unless the engineer on watch is well acquainted with all the machinery, he cannot act promptly in case of emergency, but will be compelled to send for the chief or find someone under him who can furnish detailed knowledge of the part in question. The promptness and confidence with which he can act at all times depend upon his knowledge of all the parts of the machinery.

Before Starting. Just what to do before starting depends largely upon the prevailing conditions and the arrangement of the machinery. In general, the following should be observed:

All gear used in port or for repairs should be stowed away and all covers replaced. Such valves as the inlet and the outlet valves of the circulating pump and all valves to bilge pipes should be tried and put in proper condition. The

outboard delivery valves from all pumps should receive especial attention. The valves to jackets and the bulkhead and regulating valves should be opened and inspected. The valves in the main steam pipe should not be closed tightly or they will be set fast when steam enters

The oil cups and lubricators should be examined and put in good working order and the necessary worsteds adjusted

The various joints should be inspected and the glands packed

Pressure and vacuum gages should be connected and the shut-off cocks tried

The bright parts of the machinery that are likely to become splashed with water should be oiled

Auxiliary engines should be tried by steam if possible; if not, by hand. Such auxiliaries as the steering engine, circulating engines, and the electric-lighting engines should receive careful attention In all cases, the reversing engine should be tried before using the main engines and before entering port it should again be tried to make sure that it works properly

The main engine should be oiled at all the rubbing and rotating parts

An important item is the examination of the crank pits and all the working parts. If these parts are not examined, some obstruction may prevent the engine from starting The main engines should be turned through at least one revolution, both ahead and astern, by hand

In case forced draft is used with closed stokeholds, the draft gages should be cleaned and filled with water and the air-tight doors should be examined and rigged. The fans should be carefully oiled and adjusted

To Start Engine. In starting an engine the engineer in charge must use the knowledge gained from experience, as no set rules will apply to all engines. For instance, a small single-cylinder engine is not started in the same manner as a large triple-expansion engine. In the following we will consider the types of machinery most used—the triple-expansion engine and surface condenser.

In general, to start an engine it is first necessary to warm the cylinders and form a vacuum in the condenser, the engine can then be started by admitting steam to the cylinders.

To Form Vacuum. It is usual to fit an independent circulating pump, so the Kingston or sea-valve should be opened and the discharge valve tested to see if it lifts readily. The circulating pump is then started so that the condenser will not become heated by the drains and exhaust steam. The auxiliary air-pumps should then be started to keep the main and auxiliary condensers free from water and to form a partial vacuum. If the air-pump for the main condenser is independent, it may be started so as to form a vacuum.

To Warm the Engines. To warm the engines, all cylinder, receiver, and steam chest drains are put in communication with the

condenser. In order to ascertain whether or not the drains are working properly, a by-pass arrangement is often fitted. This arrangement connects the drains to the bilges. The jackets are usually trapped to the hot well or feed tanks, but can be drained directly to the bilges. If all the drains are in order, open slightly the throttle valve and all valves in the main steam pipe. This will admit a little steam to the high-pressure steam chest. Steam is also admitted to the jackets to assist in warming the cylinders.

Now open the by-pass valves a little to admit steam to the receivers. The steam in the receivers finds its way into the cylinders and helps in the warming up. To warm both ends of the cylinders move the valve gear back and forth slowly from full gear ahead to full gear astern. The throttle may now be opened a little wider, enough to set the engine in motion. By means of the reversing gear, the cranks can be made to move back and forth without making a complete revolution.

Opening the Throttle. We will assume that the engine is thoroughly warm and (as the drains are open) free from water. Steam is in the jackets and the starting engine and starting valves ready. The centrifugal pump is at work circulating water through the condenser and either the auxiliary air-pump or an independent air-pump is at work

To start the engines, run the links into full gear ahead or astern and open the throttle valve. In case the engines do not start, use the by-pass or auxiliary starting valves. The engines should be started slowly and the speed gradually increased by admitting more steam. After the engines have made 200 revolutions or more, the drain cocks may be closed.

Causes of Failure to Start. Marine engines may fail to start from many causes, but if proper precautions are observed before trying to start there should be no difficulty. Among the causes which are not apparent from the exterior are:

The throttle valve spindle may be broken.

The high-pressure valve (if a slide valve) may be off its seat and admit steam to both ends.

The engine may be gagged; that is, the throttle will supply steam to one side of the high-pressure cylinder and the by-pass valves admit steam to the opposite side of the intermediate or low. In this

case the engine will not move, as the pressures are equalized. In using the by-pass valves, the valve or valves should be used which will produce a turning moment on the shaft. Let us suppose that both the high- and low-pressure valves cover the ports, and the intermediate slide valve is in such a position that steam can enter that cylinder. If now the throttle is opened, the engine will not start, because both ports are closed. If the by-pass valves to both receivers are opened, steam will be admitted to the proper side of the intermediate piston. Also the steam in the low-pressure receiver will find its way through the exhaust cavity of the low-pressure slide valve to the other side of the intermediate cylinder. The result will be that the engine will not start because the high and low are not available for starting and the pressures on the intermediate piston will balance. In this case steam should be admitted to the intermediate receiver only. If steam is admitted to the low-pressure receiver only, it tends to force the intermediate valve off its seat.

The opening of the wrong starting valves will frequently produce a similar situation.

If the engine has become gagged, it should be freed from steam This may be done by closing the throttle and moving the link to the opposite extreme position. The engine can then be started in this direction and then be quickly reversed; or it may be started in the proper direction if the mistake is not repeated. In case the engine will not start, one of the following conditions may be the cause:

(a) The valve stem may have become broken inside the chest or the valve may have become loose on the stem.

(b) One of the eccentrics may be broken or slipped on the shaft

(c) Bearings set up too tightly or too much compression on the packing in stuffing boxes often prevent starting.

(d) The propeller may be fouled by a rope or other obstruction.

(e) The turning gear may not be disconnected; that is, the worm may still be in gear with the worm wheel.

Adjustments After Starting. After the engine has been running for a short time, the following adjustments should be made:

The speed of the feed pumps to maintain the proper water level in the boilers.

The supply of circulating water to the condensing equipment.

The amount of circulating water around the main bearings should be reduced as low as possible to relieve the work of the bilge pumps.

The pressures in the steam jackets and the valves in the drains should be regulated.

Lubrication. The oil cups on bearings require special attention. The caps of lubricators should be kept in place on the oil cups to prevent dirt and water from entering. The lubricators should be examined frequently because the pipes and passages are likely to become clogged.

For cylinder lubricator as little oil as possible should be used, so as to keep the boilers free from grease. The lubricators used for this work are discussed and described in "Steam Engines", Part II.

Hot Bearings. There are many causes for hot bearings, the most common of which is dirt. To prevent the accumulation of dirt in the bearings, the engine room, oil cups, and pipes, should be kept clean.

Insufficient and improper lubrication will almost always cause heating. If the oil enters at the top, where the pressure is greatest, suitable oilways should be cut to allow the entrance of the oil. Another method is to lead the oil to a point of low pressure.

Other causes are improper adjustment or alignment and deficient surface. These defects lead to excessive pressure in some parts, which causes heating.

In many large, modern engines, the main bearings have the castings cored out so that water circulates through the bearing continuously, but does not come in contact with the rubbing surface. In the caps there are holes to allow the hand to feel of the bearings and to allow air to circulate. The temperature of the circulating water and the hand test indicate the condition of the bearing.

In case a bearing tends to become too warm, the amount of circulating water is increased. In extreme cases of heating, the bearing may be flooded with water, thus washing out all of the dirt and reducing the temperature. If this water douche is used, plenty of oil should be supplied and the bearing given careful attention.

It may be necessary to slack back the nuts on the caps for a short time, but they should be slacked but little or there will be pounding. Sometimes the power distribution may be temporarily altered, that is, the power given out by any one cylinder may be decreased, and the power given out by the others increased by running

the link in or out and adjusting the expansion gear. It may even be necessary to reduce the speed for a time, but this is not done unless necessary, as it causes delay.

If the bearing is discovered to be hot, the water service should not be applied, as the sudden cooling may cause fracture. In this case the engine should be slowed down or stopped and the bearing cooled with oil, sulphur, or a mixture of soft soap, water, and oil.

Bearings that are lined with white metal should receive special attention, as the white metal soon becomes plastic and melts at about 400° F.

The water douche should be used only in extreme cases and with caution, because it may cause fracture and is likely to corrode and destroy the bearings. If water must be used, the parts should be cleaned and oiled as soon as the engines stop.

Hot Rods. Piston rods and valve rods are often kept lubricated by means of a large brush, called a swab. Frequently in starting, a man with a swab is stationed to keep the rods cool. If these rods become warm because of tight glands, they may be cooled by slacking back the gland and applying water and oil by means of a swab or syringe. If the rod is hot and water is applied, one side may be cooled and shortened; the result will be a bent rod. Instead of using water, the engines should be eased. If the rod cannot be felt, a few drops of oil or water syringed on the rod will show whether or not it is hot. If hot, the water will hiss or the oil will burn and cause smoke.

As with bearings, piston rods that are packed with metal packing should receive careful attention, as the packing may run and cut the rods. The principal causes for hot rods are glands too tight or not properly packed, piston rod not in line, and insufficient lubrication.

Knocks. Bearings should be adjusted while the engines are running. If a bearing is loose, it will knock at both ends of the stroke. Usually knocks can be located by the sound or by the feeling. Knocking in the cylinder may be due to a loose or broken piston ring, piston loose on the rod, or a nut or bolt loose. If knocking occurs, open the cylinder and jacket drains to be sure it is not due to an accumulation of water. If the noise continues at various speeds, it is probably due to looseness of the piston rings. If this is the case, the ring must be re-scraped and fitted.

Jackets. The pressures in the jackets should be maintained at the desired amount. The jacket drains are led either to the condenser or to the feed tank. If led to the feed tank, the temperature of the feed water is then raised. The jackets should be well drained, as water causes a crackling noise at each stroke. The remedy is to open the drains wide and, when clear of water, regulate the drain valves by increasing the opening.

Bilges. The bilge pumps should be at work constantly while the vessel is steaming, so that water will not accumulate in the bilges or crank pits. The crank pits should not be in communication with the bilges, or the oil from the crank pits will be spread over the bilges. If the stokehold bilges empty into the engine room bilges, the bilge water should be strained on account of the fine coal in the stokeholds. Strainers should be carefully attended to, as fine coal, waste, and articles carelessly left in the bilges are likely to choke them. It is considered good practice to pump from wells formed in the bilges and covered with strainers.

Linking Up. When starting, the links are placed in full gear. When running at the required speed, the engine is linked up so that the expansive working of the steam may be utilized. The best position of the links for a given speed is determined by experience. Trial will show at what position the engine will run smoothly, economically, and without too much noise. The throttle valve should be wide open, so that steam will enter the high-pressure chest at nearly boiler pressure. If the engine is running at reduced speed, it is a good plan to link up the high-pressure engine by the use of the block in the slot of the arm on the weight shaft. This will increase the total ratio of expansion, but will not reduce the port opening of the intermediate- and low-pressure cylinders. If there is any probability of a change in speed, the engineer in charge should see that the starting engine is warmed and drained from time to time and be sure that it is ready for use. Grunting of the slide valves is sometimes stopped by running the links into full gear for a short time, then adjusting them in a slightly different position.

Marking Off Nuts. In order to have a record of adjustments and to aid in adjusting bearings, the following marks are made. At each corner of the hexagonal nut near the face that bears on the washer, a number is stamped, as shown in Fig. 69. The washer is

prevented from moving by some device. A part of the circumference of the washer is marked off in, say, 10 divisions about one-half inch apart. These divisions are then sub-divided and numbered. It is then easy to record the position of the nut by noting what number on the washer coincided with the corner of the nut. Thus 1 on $1\frac{1}{2}$ or 2 on $8\frac{1}{2}$.

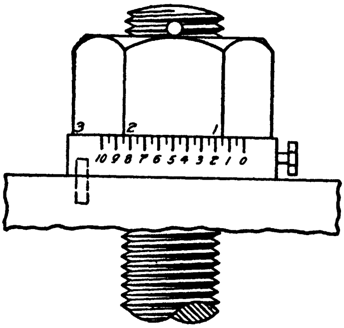


Fig 69 Marking Nuts and Washers

Refitting Bearings. To find out whether or not a bearing needs refitting and to ascertain the amount of play, a lead is taken. The cap is first removed and a piece of lead wire is laid along the journal parallel to the axis. Some engineers place two pieces around the journals near the ends and others place them diagonally. The cap is then replaced and screwed down hard on the liners. The cap is again removed and the leads taken out and examined. They should be flattened uniformly. The thickness shows the clearance. If the marks on the nuts at which the leads were taken are noted, they may be compared with the marks and leads taken sometime afterward and the location and extent of wear known.

If the leads show that the bearing needs refitting, the caps are first removed and the journal, caps, and oilways cleaned. The journal is then carefully calipered and, if found oval, cut, or rough, should be filed all over until smooth and true. This process requires considerable care and skill for the new surface must be concentric with the axis. The filed surfaces are smoothed by an oil stone or emery. If emery is used, care must be taken to clean all surfaces.

After the journals are in proper condition, the brasses, if used, are fitted by filing and scraping. A little red lead smeared on the journal will assist in the fitting. The brasses should be eased away at the sides, as the metal at those points is of no assistance, but increases the friction.

If the bearings are lined with white metal, they must be relined when the white metal is worn through. To do this a mandrel of the same size as the journal is placed in position in the bearing and the molten metal poured in or the strips of white metal are hammered

into the recesses. The metal stands clear of the brass about $\frac{1}{4}$ inch when finished.

Stopping the Vessel. *Before Entering Port.* When near port, the fires should be burning light, so that there will be no difficulty in keeping the steam pressure down. If the pressure rises when the engines are slowed down, there may be an unnecessary waste of fresh water on account of the blowing of the safety valve; the loss of fuel will also be considerable.

Before entering port all the ashes should be dumped overboard and all the water possible should be pumped out from the bilges. The reversing and capstan engines should be warmed ready for use. When the engines are slowed down, the water service should be shut off and the oil supply increased to prevent rusting of the bearings while in port. The pressures in the receivers and jackets should be watched, as they have a tendency to rise when the engines slow down.

Adjustments After Stopping. When the engines are done with, the valves in the main steam pipe and the jacket valves should be closed, but not too suddenly; the steam should then be allowed to escape from the pipe or used up by the reversing or other auxiliary engine. All drains and receiver relief valves should then be opened, and the steam should be shut off from the steering and reversing engines.

The hand-turning gear may be put in gear as soon as there is no steam left in the engine room main steam pipe. The engines should now be cleaned while warm by wiping down the rods and shafting with cotton waste and oiling the bright parts to prevent rusting.

In case the engines are stopped suddenly, notice should be immediately given in the fire room so that the draft may be checked and the evaporation reduced. If the water level is low, water should be pumped into the boilers. Every precaution should be taken to prevent an oversupply of steam, but if it is impossible to prevent the rise of pressure, the excess of steam may be used in the evaporators, distillers, etc., and in pumping out bilges and crank pits. The engines should be kept warm and well drained so as not to cause delay in starting. If the air-pump is worked by an independent engine, it should be kept working for a time, so that the condenser will not be flooded with water and injure the air-pump. If the air-pump is worked from the main engine, it will of course stop as soon as the

engines stop; in this case put on a feed-pump to keep the condenser free from water. The circulating engines may be stopped soon after the engines stop.

As in case of entering harbor, watch receiver and jacket pressures, and stop the supply of water to bearings, etc. If there is any chance of starting again soon, keep the reversing engine warm and well drained.

Precautions for Long Stay in Port. If the stay in port is to be long, the main condensers and air-pumps should be well drained and several of the boilers may be cleaned and repaired if necessary. The fires should be allowed to burn themselves out gradually. If the stop is for a short time, the fires should be banked.

Emergencies. What to do in emergencies depends upon the arrangement of the machinery. The kind and number of engines and their arrangement and capacities of the condensers and auxiliary machinery often determine what course to pursue in case any part breaks or gets out of position.

Cylinder Head Broken. If a cylinder head breaks, it should be repaired if proper means are at hand. If it cannot be repaired, the steam port which admits steam to that end may be blocked up by driving in plugs of soft pine and the engine run single-acting. This is comparatively simple if the valve is a plain slide, but with a piston valve the many ports make it more difficult. If a cylinder head of a triple-expansion engine breaks, and one engine must run single-acting, the expansion gear should be arranged so that the work will be properly divided.

Fracture in the Crankshaft. What to do in this case depends upon many conditions. If the engine is of the multicylinder type, and the crankshaft is made in interchangeable lengths, fit the spare length in place of the disabled one. In case no spare length is carried and the crankshaft of the low-pressure engine is damaged slightly, change the low-pressure length to the high-pressure engine and place the high-pressure length in place of the low. The low-pressure length transmits the most power. If the damage is considerable, such as the breaking of the crankpin, the length cannot be used and the high-pressure engine must be disconnected. If the pumps are worked from the high-pressure crosshead, repair the broken shaft, place it in the high-pressure engine, and block up the steam ports to the

high-pressure cylinder. The power is then developed in the intermediate- and low-pressure cylinders; the amount of power transmitted to the high-pressure crankshaft being just sufficient to work the pumps. Probably it will be necessary to run the engines slowly because of the weak shaft.

Piston Broken. If the piston, piston rod, or valve stem become broken and cannot be repaired, the damaged engine must be disconnected and the power furnished by the others.

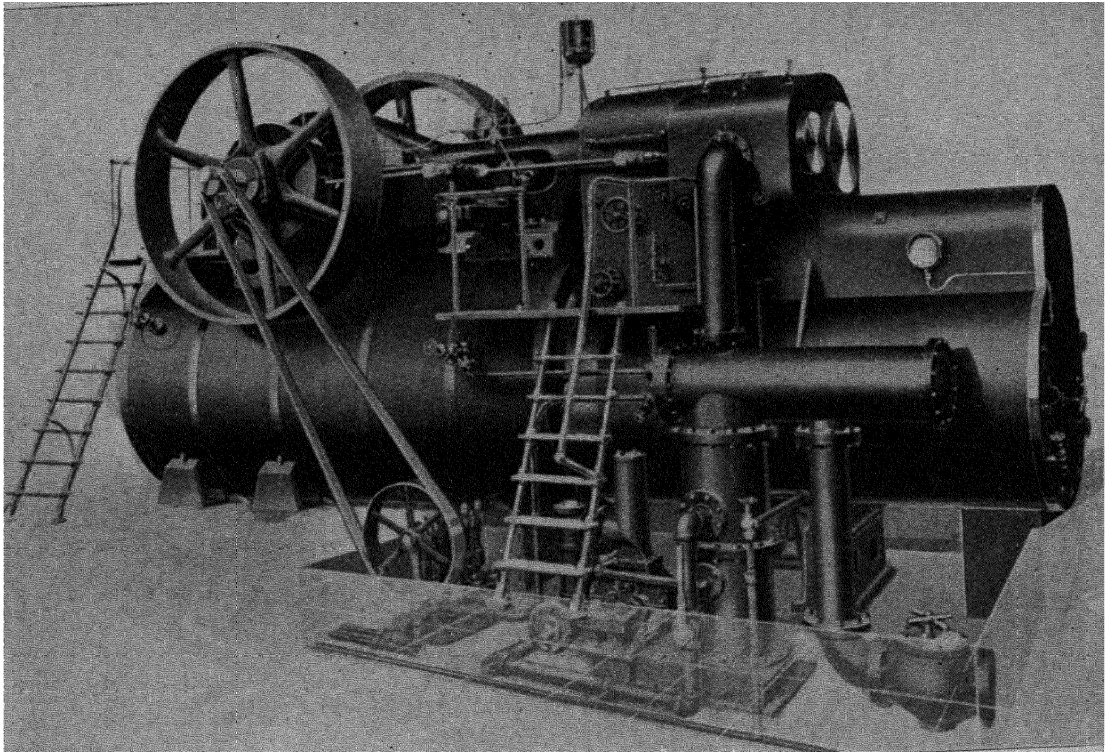
Air-Pump Broken. In case the air-pump breaks and cannot be repaired, the exhaust may be carried to the deck and the engines run non-condensing. This is a great disadvantage if the amount of fresh water carried is slight and the ship is far from port. In case no separate exhaust is possible, the auxiliary air-pumps may be connected and the ship proceed. In most cases, however, the auxiliary air-pumps are not of sufficient capacity to remove all of the condensed exhaust steam and the air; therefore, no vacuum will be carried, but the condensation may be returned to the boilers.

Bent Piston-Rod. In the case of a small rod and a long, slight bend, the rod may be straightened by placing it in a lathe and applying a powerful lever. A large rod, or one with a quick bend, should be heated to a dull red in a wood fire. The rod is then placed in a large lathe and straightened by an hydraulic jack. In doing this work care must be taken that the rod is not heated too hot, does not scale, and that the points of contact are protected by copper plates.

Eccentric Broken. If the go-ahead eccentric or eccentric rod breaks and cannot be repaired, the go-astern eccentric can be shifted in its place. The engine will now run ahead, but cannot be reversed. The go-astern end of the links must be kept from dropping by some flexible support, such as a rope or chain.

Another method is to disconnect the connecting rod from the crankpin and crosshead of the disabled engine, and block up the steam ports so that the steam will flow to the other cylinders by the shortest passage. The piston should be secured on the bottom of the cylinder. The valve should be removed. After removing the broken valve gear, the engine is ready to start. This method may be used if the pumps are worked from the low-pressure crosshead and the low-pressure engine is intact. If, however, the high-pressure eccentric is broken and the pumps are worked from that crosshead,

the same method may be pursued as described for a fractured crankshaft. That is, the valve gear should be removed, the ports blocked, and the piston, the piston rod, crosshead, and connecting rod left in place. The moving parts of the high-pressure engine will then work the pumps by means of the power transmitted to the high-pressure crank. The engine must be run slowly, but can be reversed.



SIDE VIEW OF BUCKEYE TYPE OF LOCOMOBILE SHOWING ACCESSORIES BELOW THE BOILER

Courtesy of Buckeye Engine Company, Salem, Ohio

STEAM ENGINES

PART II

MECHANICAL AND THERMAL EFFICIENCY

The brief historical review and the study of the various types of engines have served to unfold the degree of perfection that has been attained in the design and details of construction of the modern steam engine. From a mechanical standpoint, the modern engine is highly efficient. A mechanical efficiency, that is, $\frac{\text{Brake horsepower}}{\text{Indicated horsepower}}$, of from 85 to 95 per cent is not infrequently obtained. An actual test of a 12-inch$\times 19\frac{3}{4}$-inch$\times$15-inch tandem-compound Corliss engine operating non-condensing gave a mechanical efficiency of 94 per cent. That is to say, if the engine was developing 120 horsepower in the cylinders, that 112.8 horsepower would be delivered by the engine to the flywheel. In other words, the horsepower used in overcoming the friction of the various moving parts was only 7.2 or 6 per cent of the total horsepower developed.

Low Thermal Efficiency Inherent. From the standpoint of thermal efficiency, however, the modern engine is very inefficient, but it is much more efficient than the older types. Even the maximum thermal efficiency obtained is only about 15 per cent, and, under favorable conditions, this very low figure may be so reduced that the engine is operated at a great economic loss. It is now proposed to briefly point out some of the causes for the very low thermal efficiency obtained and to indicate some of the means that have been employed to increase the thermal output of the steam engine. In order to make this study it becomes necessary to again refer to steam and its properties. It is well known that steam contains a great deal of heat, and that this heat can be converted into useful work by allowing the steam to pass from the high temperature of the heat generator to the lower temperature of the refrigerator, during this change giving up heat. There are several

forms of heat engines, all of which convert the heat contained in some substance into work. The theoretically perfect engine shall be considered first, and after that the modifications that go to make up the steam engine of today.

Ideal Engine. The theoretical engine, Fig. 70, is supposed to receive heat from the generator at constant temperature T_1 until communication is interrupted at B. The working substance expands to C without losing or gaining any heat from external sources until the temperature of the refrigerator is reached. The engine now rejects heat at the constant temperature T_2 of the refrigerator and then compresses the working substance without loss or gain in the quantity of heat until the temperature of the heat generator is reached. These are ideal conditions and, if fulfilled, the efficiency of the perfect engine will depend only on the difference between the temperature at which heat is received and rejected or, in other words, it depends only upon the difference in temperature between the generator and the refrigerator.

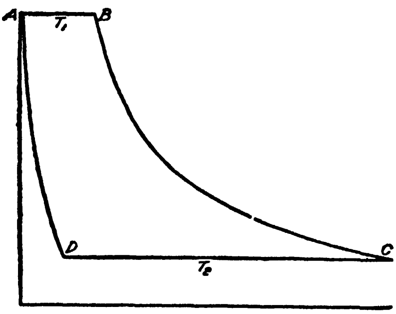


Fig 70. Theoretical Indicator Diagram

If T_1 equals the absolute temperature of the heat received and T_2 equals the absolute temperature of the heat rejected, then the thermal efficiency E of the engine will be represented by the formula

$$E=\frac{T_1-T_2}{T_1}$$

Or, in other words, the efficiency equals the absolute temperature of the heat rejected, subtracted from the absolute temperature of the heat received, and the remainder divided by the absolute temperature of the heat received.

EXAMPLE. Given an engine using steam at a 120 pounds absolute pressure, and exhausting at atmospheric pressure. What is the thermal efficiency?

SOLUTION. The absolute temperature corresponding to 120 pounds pressure is 341.31+459.5, or 800.81°, and the absolute temperature of the exhaust is 212+459.5, or 671.5°. Then

$$E = \frac{800.81 - 671.5}{800.81}$$
$$= .161, \text{ or } 16.1 \text{ per cent}$$

Losses in Practical Engine. *In General.* In actual engines this efficiency can not be realized, because the difference between the heat received and the heat rejected is not all converted into useful work. Part of it is lost by radiation, conduction, condensation, leakage, and imperfect action of the valves. The cylinder walls of the theoretical engine are supposed to be made of a non-conducting material, while in the actual engine the walls are of metal, which admits of a ready interchange of heat between cylinder and steam. This action of the walls can not be overcome and is so important that a failure to consider its influence will lead to serious errors in computations, and no design can be made intelligently if based on the theory of the engine with non-conducting walls. In theoretical engines steam expands without the loss of any heat, while in the actual engine a large amount of heat is lost by radiation. There is also a considerable loss of pressure between the boiler and the engine, due to the resistance offered by the pipes and cylinder passages. In a slow-speed engine with large and direct ports and valves this trouble is reduced to a minimum. The imperfect action of the valve gears may also be lessened with due care, but the action of the cylinder walls still remains to be overcome.

Theoretical and Actual Card Analysis. In the theoretical card, admission is at constant boiler pressure, cut-off is sharp, expansion is complete—that is, expansion continues until the temperature falls to that of the condenser and the exhaust is at condenser pressure—and the piston always sweeps the full length of the cylinder.

In the actual engine there is a considerable loss of pressure between boiler and engine, and the wire drawing of the ports and valves tends to cause a sloping steam line. Condensation at the beginning of the stroke causes the real expansion line to fall below the theoretical, while re-evaporation causes it to rise above the theoretical toward the end of expansion. In the actual engine, release takes place before the end of the stroke, expansion is not complete, that is, the pressure at release is above that of the condenser, and the resistance of exhaust ports causes the back pressure to be above the actual condenser pressure. Moreover, the piston does not sweep the

full length of the cylinder, and the clearance space must be filled with steam, which does very little work. The theoretical and actual cards are shown in Fig. 71.

Mechanical Losses. It has been shown that the efficiency of the theoretical engine is purely a thermal consideration; the efficiency of the actual engine, however, is largely a mechanical matter. The unit of work is the horsepower, which corresponds to the development of 33,000 foot pounds per minute. As 778 foot pounds are equivalent to one British Thermal Unit, 33,000 foot pounds per minute, or one horsepower, is equivalent to $33{,}000 \div 778$ or 42.42 British Thermal Units. Now if a certain engine uses 84.84 British

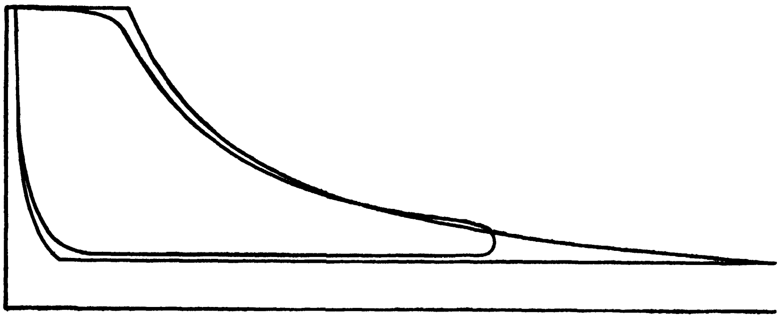


Fig. 71 Superposed Ideal and Actual Indicator Diagrams

Thermal Units per horsepower per minute, it is evident that its efficiency will only be one-half, or 50 per cent, because 42.42 is one-half of 84.84. Hence, it may be said that the efficiency of the actual

engine is equal to $\dfrac{42.42}{\text{British Thermal Units per horsepower per minute}}$

This efficiency is always much less than that of the perfect engine.

ANALYSIS OF LOSSES

The effect of some of the losses in the steam engine and the methods for decreasing them will now be considered.

Radiation. In the first place, the metal walls of the cylinder, being good conductors of heat, become heated by the steam within and transmit this heat by conduction and radiation to the air or external bodies. With the cylinder well lagged, much less heat is lost by radiation. If the lagging were perfect and the temperature

of the cylinder remained the same as the temperature of the steam throughout the stroke, there would be no loss by radiation, but heat would still be lost by conduction to the different parts of the engine.

Cooling by Expansion. During expansion, the temperature and pressure of the steam decrease as the volume increases, and the temperature at exhaust is much less than the temperature at admission. In the perfect engine, the working substance after exhaust is compressed to the temperature at admission, but in the actual engine much of this steam is lost and the compression of a part of it is incomplete, so that its temperature is less than the temperature at admission.

Steam Condensation and Re-Evaporation. Consider an engine operating with admission at 100 pounds absolute and exhaust at 18 pounds absolute. From steam tables the temperature at admission is found to be 327.86°, and at exhaust 222.4°. The metal walls of the cylinder, being good conductors and radiators of heat, are cooled by the low temperature of exhaust, so that the entering steam in passing through ports and into a cylinder is subjected to a temperature of more than 100° cooler than the steam. This means that heat must flow from the steam to the metal until both are of the same temperature. This causes the steam to give up part of its latent heat, and as saturated steam can not lose any of its heat without condensation, the cylinder walls become covered with a film of moisture, usually spoken of as initial condensation. This condensation in simple unjacketed engines, working under fair conditions, may easily be 2C per cent or more of the entering steam. The moisture in the cylinder has, of course, the same temperature as the steam; it has simply lost its heat of vaporization.

Although metal is a good conductor of heat, it can not give up or absorb heat instantly; consequently during expansion, the temperature of the steam falls more rapidly than that of the cylinder. This allows heat to flow from the cylinder walls to the moisture on them. As fast as the steam expands so that the pressure in the cylinder becomes less, this condensation will begin to evaporate. As the pressure falls it requires less and less heat to form steam and, therefore, more and more of this moisture will be evaporated. At release the pressure drops suddenly, more heat at once flows from the cylinder walls, and re-evaporation continues throughout the exhaust. Prob-

ably all of the water remaining in the cylinder at release is now re-evaporated, blows out into the air of the condenser, and is lost as far as useful work is concerned.

The steam that is first condensed in the cylinder does no work; its heat is used to warm up the cylinder, and later, when it is re-evaporated, it works only during a part of the expansion and at a reduced efficiency, because it is re-evaporated at a pressure and, consequently, at a temperature very much lower than that of admission. If the cut-off is short, perhaps 20 per cent of the steam condensed may be re-evaporated during expansion; if the cut-off is long, 10 per cent may be re-evaporated, the rest remaining in the cylinder at release, still in the form of moisture. Thus some of the entering steam passes through the cylinder as moisture until after cut-off, and still more passes entirely through without doing any work.

Suppose an engine is using 30 pounds of steam per horsepower per hour and admission is at 100 pounds absolute. The latent heat of vaporization at this pressure is 884 British Thermal Units per pound. If the condensation amounts to 33⅓ per cent, then 10 pounds are condensed and there is lost 10 times 884, or 8,840 British Thermal Units per hour, or 147.3 per minute; and since 42.42 British Thermal Units represent 1 horsepower, there is lost by condensation 147.3 divided by 42.42, or 3½ horsepower (nearly). If the cut-off is shortened, the condensation increases and may amount to 50 per cent. Of course, very much less steam is used at a short cut-off than with a long cut-off, and doubtless in many cases 50 per cent of the steam at short cut-off is not as great an absolute quantity as 30 per cent at a long cut-off.

Exhaust Waste. In addition to the actual loss from condensation in the cylinder, there is still another loss due to re-evaporation. Suppose, as before, that 10 pounds of steam are condensed in the cylinder, and that 20 per cent of this is re-evaporated during expansion. This will leave 8 pounds to be re-evaporated during exhaust. Suppose the exhaust is at 3 pounds above atmospheric pressure, or 18 pounds absolute (about). Then the heat of vaporization is 963.1 British Thermal Units per pound of steam, and it will require 8 times 963.1, or 7704.8 British Thermal Units, to evaporate the 8 pounds. All of this heat is taken from the cylinder, leaving the engine much cooler than it would be were it not for this re-evaporation. This

gives some idea of the great amount of heat passing away at exhaust, which is known as the *exhaust waste.*

Clearance. In all cylinders it is necessary to have a little space between the cylinder cover and the piston when at the end of the stroke. In vertical engines the space is greater at the bottom than at the top. The volume of this space, together with the volume of the steam ports, is called the clearance. It varies from 1 to about 15 per cent, depending upon the type and speed of the engine—the higher the speed, the greater the clearance. This clearance space must be filled with steam before the piston receives full pressure; and the volume of the clearance offers additional surface for condensation.

Friction. Another important loss is that due to friction. It is well known that it takes considerable power to move an unloaded engine; if fitted with a plain, unbalanced slide valve, the power necessary to move the valve alone is considerable. The piston is made steam-tight by packing rings, and leakage around the piston rod is prevented by stuffing boxes. All these devices cause friction as well as wear at the joints. The amount of power wasted in friction varies greatly, depending upon the kind of valves, general workmanship, state of repair, and lubrication.

OPERATION ECONOMIES

The foregoing discussion has served to indicate that the larger part of the heat loss occurring in the steam engine is due to initial condensation, exhaust waste, and clearance, although the effect of the latter has been greatly reduced by improvement in design. Regarding the methods devised for reducing the amount of initial condensation, the high-speed engine has in a measure decreased this difficulty because of the very high piston speed employed. Since the piston speeds are high, the length of time the steam remains in the cylinder has been greatly lessened; hence the transference of heat is considerably reduced. The piston speed is limited, however, by the performance of the valve gear, it being well known that the most efficient valve gears are those employed on the low-speed engines. Increased piston speed also calls for more clearance space, hence the possible gain in economy from high piston speed is limited by the performance of the valve gear and the clearance required for the higher speeds.

The application of the idea of multiple expansion, or compounding, has materially reduced the losses both by lessening the amount of condensation and also by utilizing the re-evaporated steam and the steam that leaks by the piston, which in some cases may be considerable, and this important improvement will be discussed first. In addition, other means have been employed for the purpose of increasing the economic performance of the steam engine, as for instance, *jacketing*, *superheating*, and the *use of condensers*.

MULTIPLE EXPANSION

Two engines may be used together on the same shaft, partly expanding the steam in one of the cylinders and then passing it over to the other to finish the expansion. One advantage from this arrangement is that the parts can be made lighter. The high-pressure cylinder can be of much less diameter than would be possible if the entire expansion were to take place in one cylinder. This, of course, makes the pressure exerted on the piston rod much less, and the piston rod and connecting rod can thus be made much lighter. The low-pressure cylinder must be larger than it otherwise would be, but its parts need not be much heavier, because the pressure per square inch is always low.

This arrangement gives not only the advantage of lighter parts, but a decided increase of economy over the single-cylinder type. If attention is given to the matter, a loss of economy would be expected, because the steam is exposed to a much larger surface through which to lose heat, but the gain comes from another source and is sufficient to entirely counterbalance the effect of a larger cylinder surface.

Less Condensation. When very high pressure steam and a large ratio of expansion is used, the difference between the temperature of the entering and of the exhaust steam is great. For instance, suppose steam at 160 pounds (gauge) pressure enters the cylinders and the exhaust pressure is 2 pounds (gauge), the difference in temperature as taken from steam tables is 370.7° − 218.2°, or 152.5°. This difference becomes nearly 230 degrees if the steam is condensed to about three pounds absolute pressure. The cylinder and ports of the engine are cooled to the low temperature of the exhaust steam and, as we have seen, a considerable quantity of the entering steam

is condensed to give up heat enough to raise the temperature of the cylinder to that of the entering steam. As the ratio of expansion increases, the difference in temperature increases, and consequently the amount of steam thus condensed also increases. To keep this initial condensation as small as possible, the range of temperature must be limited, that is, it must not have as great a difference between admission and exhaust. To do this the expansion of the steam must be divided between two or more cylinders.

It will be remembered that the great trouble Watt found with Newcomen's engine was its great amount of condensation, and he stated as the law which all engines should try to approach, that *the cylinder should be kept as hot as the steam which enters it.* This is to avoid condensation when steam first enters. If, instead of expanding the steam in one cylinder, it be expanded partly in one and then finished in another, it will have passed out of the first cylinder before its temperature has dropped a great deal, and consequently the cylinder walls will be hotter than they would have been if the expansion had taken place entirely in one cylinder. This would then reduce the amount of steam condensed. The importance of this may not be evident at first, but it makes a great difference in the economy of the engine. If there is less condensation, there will be less moisture to re-evaporate, and consequently less exhaust waste, hence there will be a saving in two ways.

Methods of Compounding. In a compound engine the steam is first admitted to the smaller, or high-pressure, cylinder and then exhausted into the larger, or low-pressure, cylinder.

Suppose steam at 160 pounds (gauge) pressure is admitted to a cylinder, and the ratio of expansion is such that the steam is exhausted at about 60 pounds (gauge) pressure; then the difference of temperature is 370.7° − 307.4°, or 63.3°.

If now the steam when exhausted from the first cylinder enters a second and is allowed to complete its expansion, so that the exhaust pressure is about two pounds (gauge) pressure, the difference of temperature in the cylinder will be 307.4° − 218.2°, or 89.2°.

Then for the simple engine, if the exhaust pressure is two pounds (gauge), the difference of temperature is 152.5 degrees, while in the compound engine this difference is divided into two parts, 63.3 degrees and 89.2 degrees. The cylinder condensation for both

cylinders of the compound engine will be much less than if the total expansion took place in a single cylinder. The cylinders should be so proportioned that the same quantity of work may be done in each.

If there are two stages of expansion, the engine is called simply *compound;* three stages, *triple;* and four, *quadruple.*

Exhaust Waste Utilized. Besides reducing the excessive condensation, there is still another gain in using multiple expansion. It has been shown how much heat is lost by the exhaust waste, which in the simple engine blows into the air or into the condenser and is entirely lost. In the multiple-expansion engine the exhaust and re-evaporation from one cylinder passes into the next and does work there; furthermore, any leakage from the high-pressure cylinder is also allowed to do work in the low-pressure cylinder.

JACKETING

The most primitive method of effecting steam economy is by jacketing, which principle Watt early recognized and adopted. This method reduces the loss due to cylinder condensation by supplying heat to the steam while it is in the cylinder, that is, by surrounding the cylinder with an iron casting and allowing live steam to circulate in the annular space thus formed. The cylinder covers are also made hollow to permit a circulation of live steam. A cylinder having the annular space A, Fig. 72, filled with steam is said to be jacketed. A lining L is often used in jacketed cylinders.

Function of Jacket. The function of the jacket is to supply heat to the cylinder walls to make up for that abstracted during expansion and exhaust, so that at admission the cylinder will be as hot as possible. The result is, that the difference in temperature between the cylinder walls and the entering steam is considerably less than in engines where no jacket is used. Condensation is therefore reduced and, since heat flows from the jacket to the cylinder during expansion, a much larger amount of this condensation is re-evaporated before release and it thus has an opportunity to do some work in the cylinder. This leaves a comparatively small amount of exhaust waste and the heat thus abstracted is made up from the steam in the jacket. Since a large amount of heat is given up by the jacket steam, a good deal of it must be condensed. Thus the question is asked: "What is the advantage of this method over

that of allowing the entering steam to supply the heat by its own condensation?" This question is answered briefly as follows:

The loss of heat by condensing the steam would be less if the inside of the cylinder could be kept dry. It has been indicated how the moisture that collects by condensation is re-evaporated during expansion and exhaust because the pressure falls and the cylinder walls are hotter than the steam. This re-evaporation takes place at the expense of the heat in the cylinder walls and they are thus cooled. It has already been shown that a great many British Thermal Units

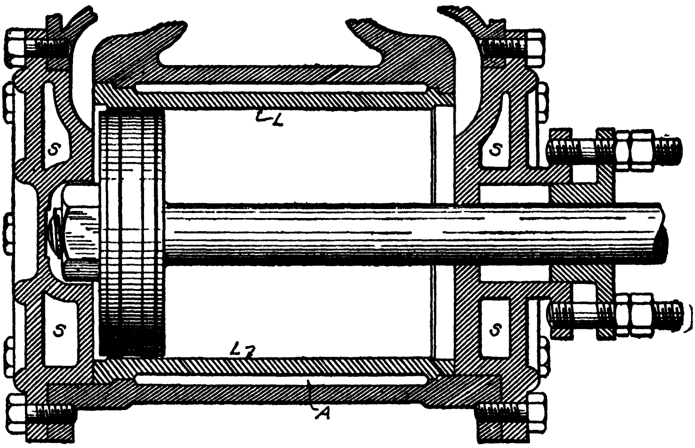


Fig 72 Section of Steam Engine Cylinder, Showing Method of Jacketing

are thus taken from the cylinder and thrown out at exhaust at every stroke. Now if the inside of the cylinder can be kept dry so that there will be little or no re-evaporation at exhaust, it will cause a considerable saving. The steam that condenses in the jacket does not re-evaporate in it; but is returned to the boiler as feed water, so that the only heat lost is the latent heat given up during condensation. If the cylinder is heated from within, both the latent heat given up by condensation and the latent heat required for re-evaporation are lost.

In a triple-expansion engine there is one distinct advantage in allowing condensation in the cylinder, for this moisture acts as a lubricant, and as the heat of re-evaporation passes into the next cylinder and there does work, there is very little loss.

Saving Due to Jacketing. It is evident that a large part of the heat of the steam jacket flows to the cylinder during exhaust and is thus entirely lost in the simple engine. In the triple engine, however, this heat passes into the intermediate and low-pressure cylinders; consequently we might expect a greater gain from using a jacket on a triple engine than on a large, simple engine. The main advantage of the jacket has been previously pointed out, and as in all cases the gain is small, there is to be found a considerable diversity of opinion as to its real advantages. On some engines there is undoubtedly little if any gain, the largest gain being in the smaller engines of, say, 200 horsepower and under. On very small engines, such as a 5-inch×10-inch engine when developing only one and one-half horsepower under light load, the gain is as much as 30 per cent. On a 10-horsepower engine the gain might be as much as 25 per cent, while on engines of about 200 horsepower the gain would probably be 5 to 10 per cent for simple condensing and compound condensing, and from 10 to 15 per cent for triple expansion. The saving on large engines of, say, 1,000 horsepower is very small, the reason being that large engines offer less cylinder surface per unit of volume than small ones, and hence have proportionately less cylinder condensation. The very small engines, in which the gain would be the greatest, are seldom jacketed, because they are built for inexpensive machines and the first cost is of more consequence than the economy of operation. Owing to the cost of construction and the care necessary to keep jackets operative, the use of the jacket has gradually diminished. Furthermore, the introduction of the high-speed and compound engines, as well as the use of superheated steam, has reduced the advantage of jacketing to relative insignificance.

SUPERHEATING

General Practice. The use of superheated steam is rather a modern practice, although for many years previous to its adoption engineers had appreciated its value in producing steam engine economy. The reason for its delayed adoption in a practical way was due to the mechanical difficulties met with in superheating the steam and also to the increased cost of maintenance produced by its use. In recent years both of the objectionable features above mentioned have been, in a large measure, overcome, so that today

superheat is being used in a large number of power plants, and also in steam locomotives.

Before describing a superheater, it may perhaps be well to clearly define what is meant by superheated steam. Water, when confined in a vessel and heated sufficiently, turns into steam, which, if some water still remains, is spoken of as saturated steam. Saturated steam when further heated becomes superheated steam, if it is separated from the water. To bring about this separation, a superheater is necessary. Superheaters vary considerably in details of construction according to the service for which they are designed,

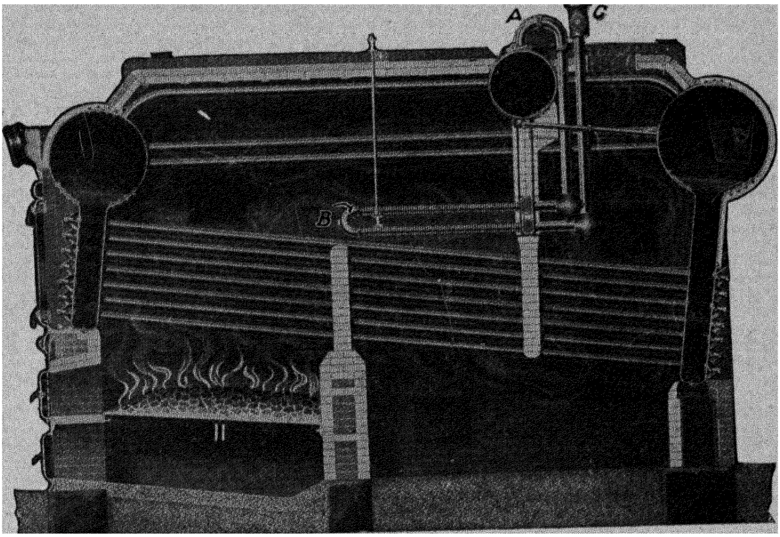


Fig 73 Section of Water-Tube Boiler Showing Application of Foster Superheater

there being, for instance, quite a difference between the superheater designed for a stationary plant and one designed for a locomotive.

Foster Superheater. A Foster superheater as applied to a water-tube boiler is illustrated in Fig. 73. The superheating element is shown at *B*, which is connected to the steam space of the boiler by the pipe *A*. The saturated steam from the boiler passes through the pipe *A*, through the superheater, and then is conveyed to the engine through the valve *C*. In this installation the superheater is placed in the passage provided for the transmission gases to the chimney, hence it is heated by what would otherwise be lost heat. The manner of installing superheaters varies a great deal. Some are

entirely separated from the boiler, being self-contained and supplied with a grate for separate firing.

The Foster superheater, Figs. 73 and 74, is made up of a number of elements placed parallel to each other, each of which consists of two straight steel tubes, one inside of the other. The elements are joined at one end to manifolds or connecting headers, and at the other end to return headers for which return bends are often substituted. On the outside of the tubes *B*, Fig. 74, are fitted a series of cast-iron annular flanges *D*, placed close to each other and carefully fitted to the tube so as to be practically integral with it, at the same time exposing an external surface of cast iron, which metal is best adapted to resist the action of the heated gases. The rings are carefully bored to gauge, and shrunk on the tubes. Once being in position, the rings and tubes act virtually as a unit. As the coefficient of expansion of steel is a trifle greater than that of cast iron, the rings grip the tubes even tighter when in service. This form of construction is flexible and durable. It provides a section of great strength and entire freedom from internal strains. The mass of metal in the tubes and covering acts as a reservoir for heat, which is imparted to the steam evenly, tending to secure a constant temperature of steam, even though the temperature of the hot gases does fluctuate. The seamless drawn tube secures great initial strength, which is reinforced by the rings shrunk on the outside. Inside of the elements there are placed other tubes *C* of wrought iron, which are centrally supported by means of knobs or buttons regularly spaced throughout their length. These inner tubes are closed at the ends. A thin annular passage *E* from the steam is thus formed between the inner and the outer tubes. The steam clinging closely to the heating surface is quickly heated in the most efficient manner.

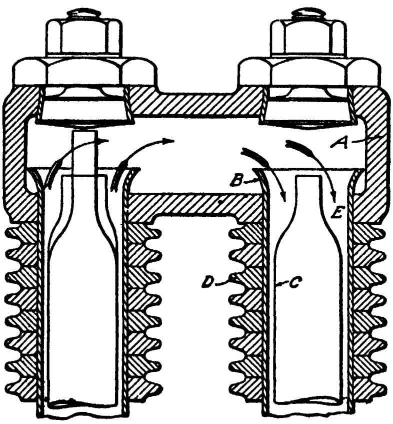


Fig 74 Section of Foster Superheater Tubes

The superheater must be as free as possible from the liability of burning out in case of a chance of overheating of the exposed

surfaces. The circulation must be properly distributed throughout the superheater at full load as well as at partial loads. The various parts must be accessible for inspection, both externally and internally, and must be readily renewable or easily repaired. There must be provision for free expansion and contraction of the various parts. The supporting arrangement must be carefully worked out. Cast iron has given excellent results in producing durable superheaters

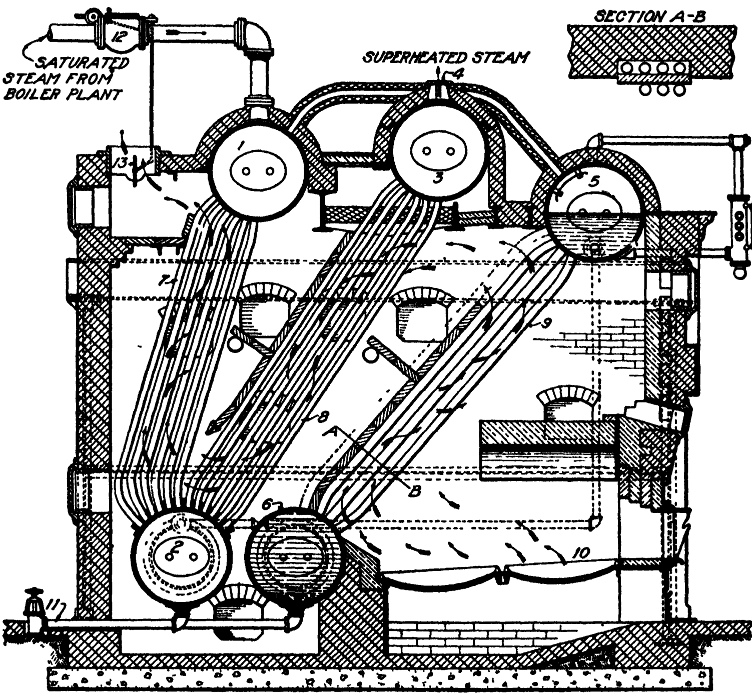


Fig. 75 Section of a Separately-Fired Type of Superheater

and has been extensively used because of its ability to withstand high temperatures. For high steam pressures, however, cast iron is not considered safe and has given way to the use of seamless steel tubes, which are homogeneous and strong, but lack the heat-resisting qualities of cast iron. It is evident that a combination of these two metals will preserve the good qualities of both.

Separately-Fired Superheater. A type of superheater differing radically from the one previously described is illustrated in Fig. 75. It is a separately-fired superheater and its construction is very

similar to the Stirling water-tube boiler. The saturated steam from the main boiler plant enters the rear superheater drum *1*, passes through the rear bank of tubes *7* into the lower drum *2*, thence to the upper drum *3*, from which it passes into the pipe line through the opening *4*. The furnace is similar to that used in the standard design of Stirling boiler. To protect the superheater tubes from high temperatures of the furnace, a sufficient amount of boiler heating surface, as drums *5* and *6* and bank of tubes *9*, is located in front of the superheater proper in order to reduce the temperature of the gases to about 1,500 degrees by the time they reach the superheater. The builders state that when the gas temperature reaches 1,500 degrees in the standard boiler, 19 per cent of the boiler heating surface has been swept over by the gases, 50 per cent of the steam produced by the boiler has been generated, and the boiler heating surface per horsepower is 3.8 square feet. Consequently in the independently fired superheater shown in Fig. 75, 50 per cent of the heat absorbed is used to generate the steam, which is added to steam furnished by the main boiler plant and hence increases the capacity of the plant in proportion. The remaining 50 per cent of the heat, a portion of which passes out the stack, is absorbed by the superheater and superheats both the steam from the main boiler plant and that from the front bank of water tubes. The superheater, because of the front generator set, will produce about 12 per cent of the amount of steam furnished by the main boiler plant. As a further precaution against any possible overheating of the superheater tubes near the furnace, a flap valve *12* is placed in the pipe conveying saturated steam to the superheater, as shown in Fig. 75. The spindle of this valve is connected by links to the superheater damper *13*, so that the damper to the opening is regulated according to the quantity of steam flowing into the superheater. If the steam flow stops, the valve *12* drops to its seat and the damper *13* is closed. Independently fired superheaters are furnished in any desired capacity, suitable for any degree of superheat up to about 300° F. The upper water drum *5* and the lower superheater drum *2* are connected by piping, hence, if desired, the superheater sections may be flooded, converting the whole into a saturated steam boiler.

Purposes of Superheaters. These two types of superheaters illustrated and described will suffice, and we may now direct our

attention to a study of the purposes of the superheater and to some consideration of the economy secured by its use. The purposes of superheating steam, as practiced in the past and as recognized at present, are, according to Thurston, the following:

(1) Raising the temperature which constitutes the upper limit in the operation of the heat-engine in such a manner as to increase the thermodynamic efficiency of the working fluid.

(2) To so surcharge the steam with heat that it may surrender as much as may be required to prevent initial condensation at entrance into cylinder and still perform the work of expansion without condensation or serious cooling of the surrounding walls of the cylinder.

(3) To make the weight of the steam entering the condenser and its final heat charge a minimum, with a view to the reduction of the volume of the condensing water and of the magnitude and cost of the air pump and condenser system to a minimum.

(4) To reduce the back pressure and thus to increase the power developed from a given charge of steam and efficiency of the engine.

(5) To increase the efficiency of the boilers both by the reduction of the quantity of the steam demanded from the original heating surface and by increasing the area of the heating surface employed to absorb the heat of the furnace and flue gases, and also by evading the waste consequent upon the production of wet steam.

If the steam entering a cylinder is only superheated enough to give dry saturated steam at cut-off, the range of temperature $\frac{T_1 - T_2}{T_1}$ of the Carnot cycle is interchanged and there is, therefore, no increase of economy from item 1. The other four sources of economy depend upon one fundamental fact—the poor conductivity of dry steam. To the property of non-conductivity of heat of superheated steam is due its great advantage. On entering a cool cylinder it slowly gives up its heat, and if the degree of superheat is sufficient there will be little or no initial condensation. The degree to which steam should be superheated is still a debated point, some engineers contending that only a very moderate degree of superheat of about 100 degrees is sufficient, whereas others maintain that no real economy is obtained with less than 200 degrees or over. When a high degree of superheat was first used, difficulties were encountered such as

the disintegration of the valves, valve seats, packing rings, and other parts subjected to the action of the superheated steam. Lubrication was also interfered with, since many of the oils used were not suited for such high temperatures. All of these difficulties no doubt account for the one time widespread objection to high degrees of superheat, but in recent years they have in a large measure been overcome. The author is familiar with the performance of a simple slide valve locomotive which has been in operation for several years under degrees of superheat ranging from 80 degrees to 214 degrees, during which time no trouble has been experienced with the valves or with the lubrication. Many European locomotives have been satisfactorily operated with high degrees of superheat, which insures the passage of steam through the cylinder with but little or no condensation.

Economic Advantages. The economy obtained by the use of superheat has been clearly demonstrated by a large number of practical tests both upon stationary engines and upon locomotives. It is to be noted also, that about the same per cent of economy has been obtained on the various types of engines tested, the stationary tests corroborating the results obtained upon the locomotive and *vice versa.* The various tests indicate a saving of from 12 to 15 per cent of the amount of steam used by the engine per indicated horsepower per hour, and a saving of coal from 20 to 25 per cent. Another very significant thing that has been determined is that the output of power has been increased from 20 to 30 per cent, depending upon the conditions. These three items of saving have hastened the installment of a large number of superheaters, so that at the present time thousands of locomotives in Europe are equipped with superheaters, and in the United States and Canada over 1,500 locomotives are so equipped. It seems that the railroads have been quicker to take up the idea of installing superheaters than have other industries, so that not nearly so many superheaters are found in stationary service.

It is to be noted that the greatest gains from the use of superheaters are to be expected in the more uneconomical plants. That is, the per cent of saving by the use of superheated steam in a simple engine would be greater than for a compound engine, and for a compound engine as compared with a triple-expansion engine. Several prominent engineers have advised the reduction of steam pressures

with a relative increase in diameter of cylinders and the use of superheated steam. The combination of a simple engine with low steam pressure and superheated steam will give an increased output of power at a small cost, a result desired by all operators.

CONDENSERS

When low-pressure steam is cooled, it gives up its latent heat, that is, it changes from a vapor to a liquid, and, as a liquid occupies much less space than an equal weight of its vapor, the changing of the steam to water greatly reduces the pressure. Therefore, by cooling the steam in an engine cylinder in front of the piston, the back pressure, or resistance, is reduced, which, in turn, reduces the pressure necessary to push the piston through the stroke and, therefore, lessens the steam required to do the work. This cooling is accomplished by some form of condenser.

Theory of Condenser Action. *Back Pressure.* In the ordinary non-condensing engine, steam can not be exhausted below a pressure of 14.7 pounds absolute, because the atmosphere exerts that amount of pressure at the opening of the exhaust pipe. In fact, this 14.7 pounds is the theoretical limit only, and in practice the exhaust is always a little above this because of resistance in the exhaust ports and exhaust pipe; so that 17 or even 18 pounds absolute back pressure is more nearly the conditions of actual service.

During the forward stroke, steam expands from the pressure at admission to a much lower pressure at release; then the valve opens for the return stroke giving full steam pressure on one side of the piston and the pressure of exhaust on the other side, the latter acting against the piston and against the force of the incoming steam. If all of this back pressure could be removed so that there would be a vacuum on the exhaust side of the piston, the power of the engine would be increased by just so many pounds of mean effective pressure, and in addition to this the steam could expand to a very much lower pressure and therefore work with greater economy.

Effect of Condensation. One pound of steam at 17 pounds absolute pressure occupies 23.38 cubic feet of space in the cylinder of the engine, but one pound of water in the condenser occupies only about 0.016 cubic feet, which makes the steam occupy nearly 1,462 times as much space as the water into which it condenses. If then,

the exhaust steam could be condensed instantly, the back pressure would be reduced almost to zero and the engine would exhaust into a vacuum.

Unfortunately the mere condensation of the steam will not give a perfect vacuum because of the air, always present in the water which comes over from the boiler. Moreover, the condensed water is hot, and the vapor rising from it in the condensing chamber, together with the air and some leakage would spoil the vacuum were

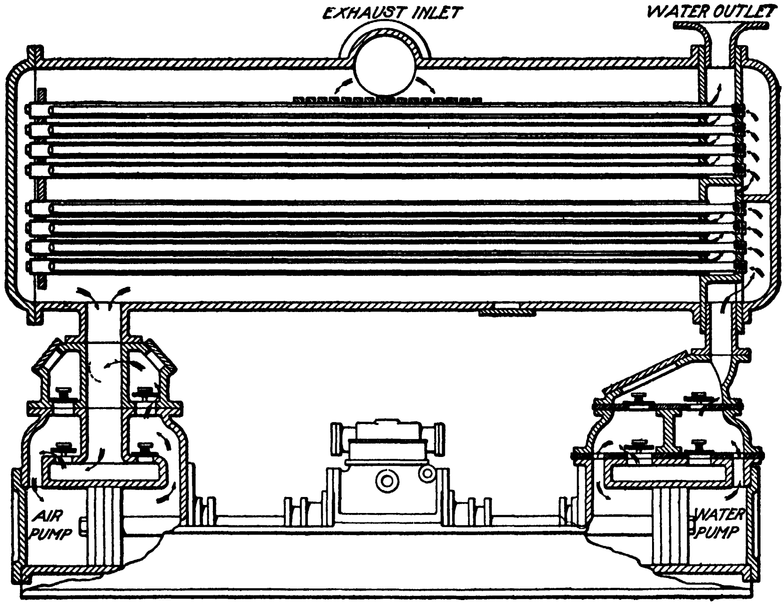


Fig 76 Section of Steam Condenser of the Surface Type

it not for the air pump, which removes the air and condensed steam. Even with the best air pump it would be impossible to maintain a perfect vacuum, but a vacuum of 26 inches, which corresponds to about 2 pounds absolute pressure, can readily be maintained in good practice.

It is well known that a certain amount of heat is required to change one pound of water at a given temperature into steam at the same temperature; this is called the latent heat of vaporization. If the steam condenses, it must give up this latent heat. The easiest ways of doing this are either to let the steam come in contact with

pipes through which cold water is circulated, as in a surface condenser, or mingle with a spray of water, as in a jet condenser. These two types will now be discussed.

Types of Condensers. Condensers may be divided into two general classes as follows:

(1) Surface condensers in which the cooling water is separated from the steam, usually by metallic surfaces in the form of tubes, the cooling water circulating on one side of this surface and the steam coming in contact with the metal on the other side.

(2) Jet condensers, including barometric condensers, siphon

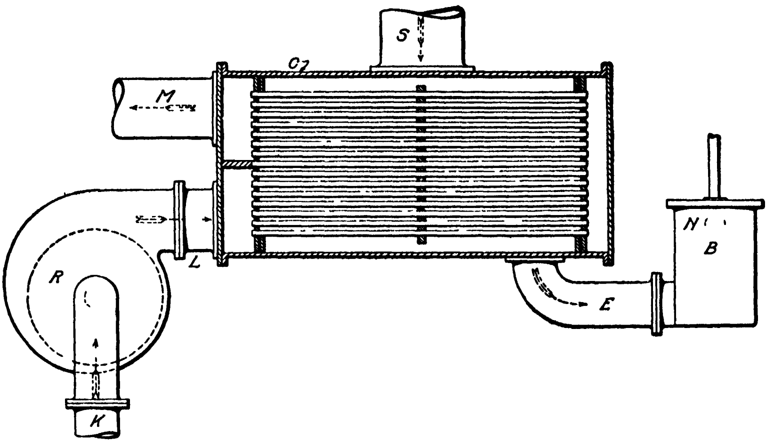


Fig 77 Diagram Showing Relation of Surface Condenser to the Pumps Necessary for Proper Operation

condensers, ejector condensers, etc., in which the cooling water mingles with the steam to be condensed.

Surface Type. The condenser shown in section in Fig. 76 is one form of the surface type, in which the air pump and the circulating pump are both direct acting and both operated by the same steam cylinder. The cool condensing water is drawn from the supply into the circulating or water pump and is forced up through the valves and water inlet to the condenser. It flows, as indicated by the arrows, through the inner tubes of the lower section, then back through the space between the inner and the outer tubes. The water then passes upward and through the upper section, as it did in the lower, then it passes out of the condenser through the water outlet, taking with it the heat it has received from the steam.

The exhaust steam from the engine enters at the exhaust inlet and comes in contact with the perforated plate, which causes it to spread. The steam expanding in the condenser comes in contact with the tubes, through which cool water is circulating, and condenses. The air pump draws the air and condensed steam out of the condenser and thus maintains a partial vacuum. This causes the exhaust steam in the engine cylinder to be drawn into the condenser, at the bottom of which it collects as it condenses and is drawn into the air pump cylinder and discharged while heated to the hot well of the boiler. The use of this hot water as feed water effects a considerable saving, but the great advantage of the condenser is the reduction of the back pressure.

Hot water can not be used by an ordinary pump as easily as cold water because of the pressure of the vapor which arises from the hot water. In the condenser shown, the water and air pumps are run by the piston in the steam cylinder. Sometimes these pumps are connected to the main engine and receive motion from the shaft or crosshead.

The general arrangement of the surface condenser with the necessary pumps is shown in Fig. 77. The cooling water enters through the pipe *K* and flows to the circulating pump *R*, which forces the water into the condenser through the pipe *L*. In case the water enters the condenser under pressure from city mains, no circulating pump is necessary. After flowing through the tubes it leaves the condenser by means of the exit *M* and flows away. Exhaust steam enters at *S* and is condensed by coming in contact with the cold tubes; the water (condensed steam) then falls to the bottom of the condenser and flows to the air pump *B* by the pipe *E*. The air pump removes the air, vapor, and condensed steam from the condenser and forces it through the pipe *N* into the hot well, from which it goes to the boilers or to the feed tank.

Circulating Pump. The circulating pump, when separate from the condenser, is usually of the centrifugal type. This pump consists of a fan or wheel which is made up of a central web (or hub) and arms (or vanes). The vanes are curved and as the water is drawn in at the central part, the vanes throw it off at the circumference. A suitable casing directs the flow. This type of pump is advantageous because there are no valves to get out of order and, as the lift

is little, if any, the pump will discharge a large volume of water in a nearly constant stream. The circulating pump is usually so placed that the water flows to it under a slight head. The pump is driven by an independent engine so that the circulating water may cool the condenser even if the main engine is not working.

Jet Type. Fig. 78 illustrates the longitudinal section of an independent jet condenser and pump. The cold water used to condense the steam enters at *A*, passes down the spray pipe *B*, and is broken into a fine spray by means of the spray cone *C*. This action insures a rapid and thorough mixing of the steam and water and consequently a rapid condensation. The exhaust steam enters at *D* with a comparatively high velocity, which is imparted to the water. The whole mixture of water, steam, and vapor passes at high velocity through the conical chamber *E* to the pump cylinder *F*, where it is forced into the pipe *G*. The spray cone is adjusted by means of the stem which passes through the stuffing box at the top of the condenser. The valves are shown at *H* and *K*. The steam end of the pump is at *L*.

In Fig. 79 a jet condenser is shown connected to a stationary engine. The exhaust pipe leads from the engine to the condenser,

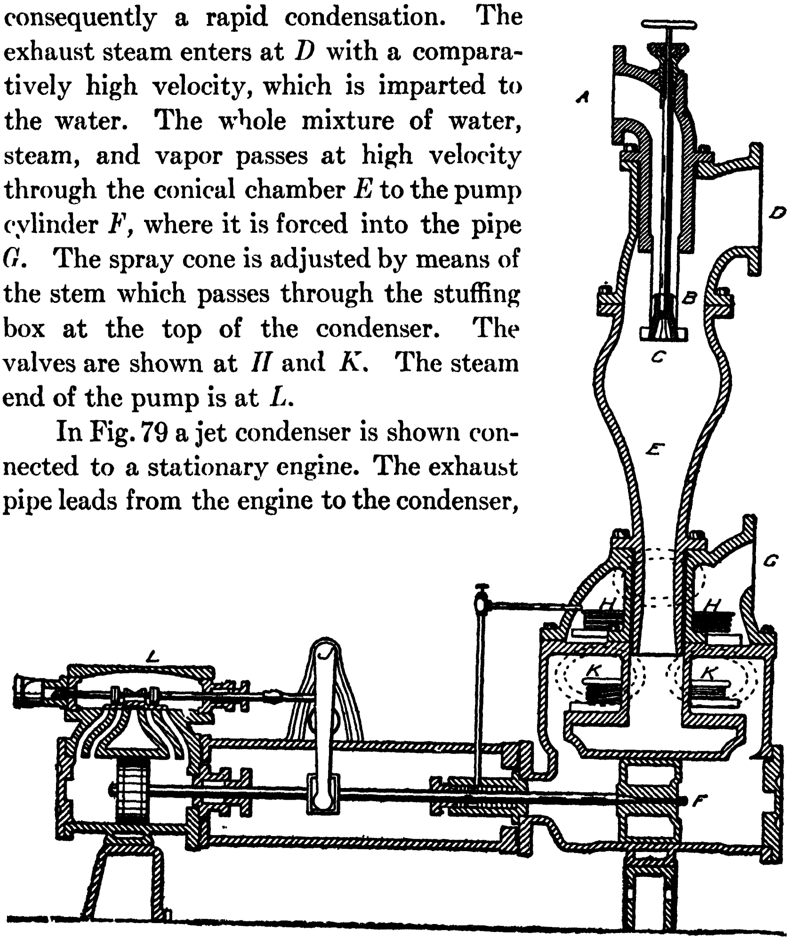


Fig. 78. Longitudinal Section of Independent Jet Condenser and Pump

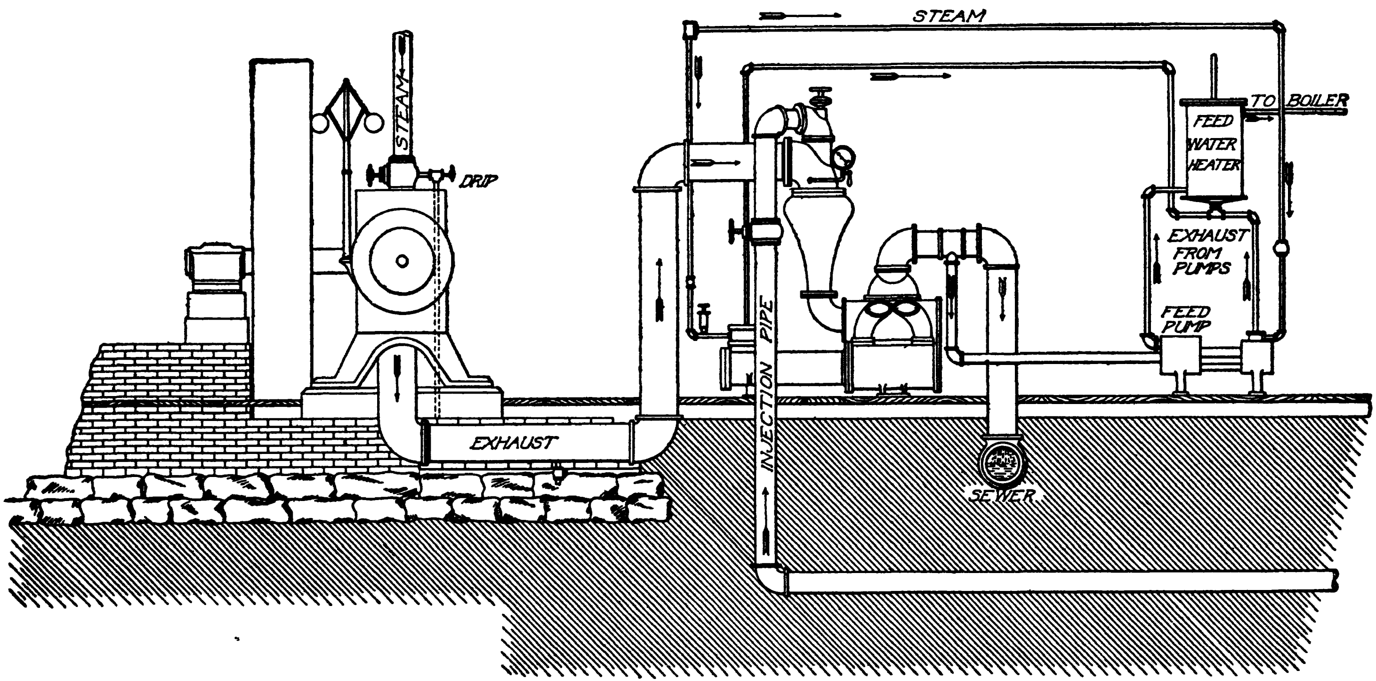


Fig 79. Diagram Showing Stationary Engine with Connection to Jet Condenser, and Other Necessary Appliances

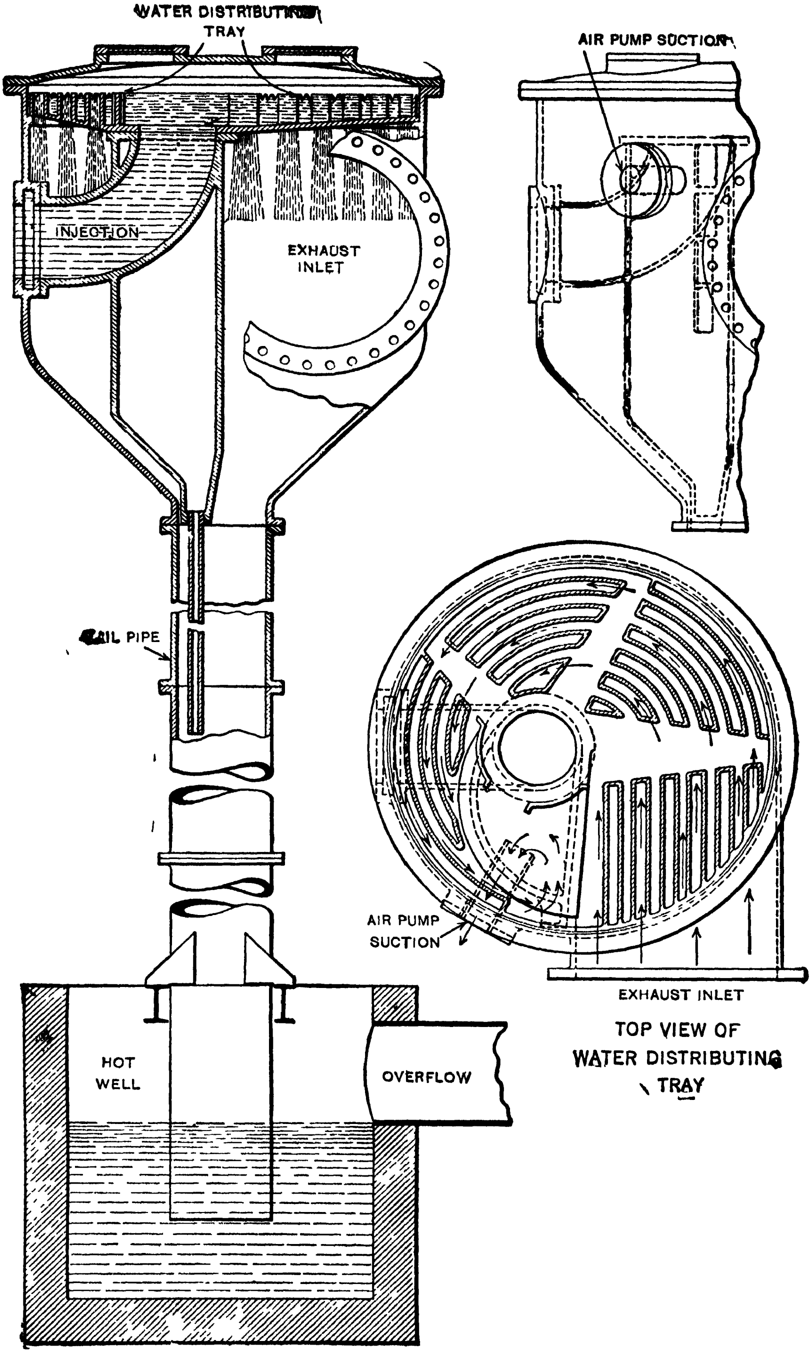


Fig 80 Alberger Barometric Jet Condenser
Courtesy of Alberger Pump and Condenser Company, New York City

the arrows indicating the direction of the flow. Cold water enters the condenser through the pipe shown. Part of the mixture of exhaust steam and condensed water goes to the feed-water heater, which is kept nearly full; the rest passes to the sewer. The heater is placed a little above the feed pump, in order that the water may enter the pump under a slight head. This is necessary because the pump can not raise water which has been warmed by exhaust steam as readily as cold water.

Barometric Condenser. A type of condenser much used with reciprocating engines, and to a limited extent with steam turbines, is the barometric condenser, shown in Fig. 80. This condenser is one of the jet type. Steam enters at the point marked "exhaust inlet" in the left-hand figure and completely fills the exhaust steam chamber, while the condensing water enters through the injection pipe. The water rises into a distributing tray where it is broken up into many finely divided streams as seen in the left-hand figure. This spray condenses the steam in the exhaust chamber and passes down the tail pipe, carrying the condensed steam with it to the hot well. Air entering with the exhaust steam is cooled and collected in the air collector inside of the condensing chamber. A vacuum is maintained in the upper part of the condenser so that any air which has been collected during the process of condensing the steam is carried away through the pipe marked "air pump suction". A small amount of the cooler injection water is allowed to mix with this air so as to cool it before it passes on to the air pumps.

Westinghouse Leblanc Condenser. With the ordinary type of reciprocating engine, a vacuum of 26 to 27 inches is usually all that is desired. With modern steam turbines, however, a vacuum of 28 to 29 inches is common practice, and in many plants even these figures are exceeded. These figures, however, cannot be attained unless a very efficient air pump is used. The Leblanc condenser is considered one of the most efficient types of the many forms of jet condensers.

Fig. 81 shows a cross section of the Leblanc jet condenser, as manufactured by the Westinghouse Machine Company. This type is especially used in the larger steam turbine installations. In this condenser steam enters through the large opening E at the top,

and the cooling water through *B*. This water is carried all around the circumference of the top of the condenser by the annular chamber *C* and is drawn inside the cone *P* through helical spray nozzles *D*, by the vacuum in the condenser. Inside of the cone *P* the water

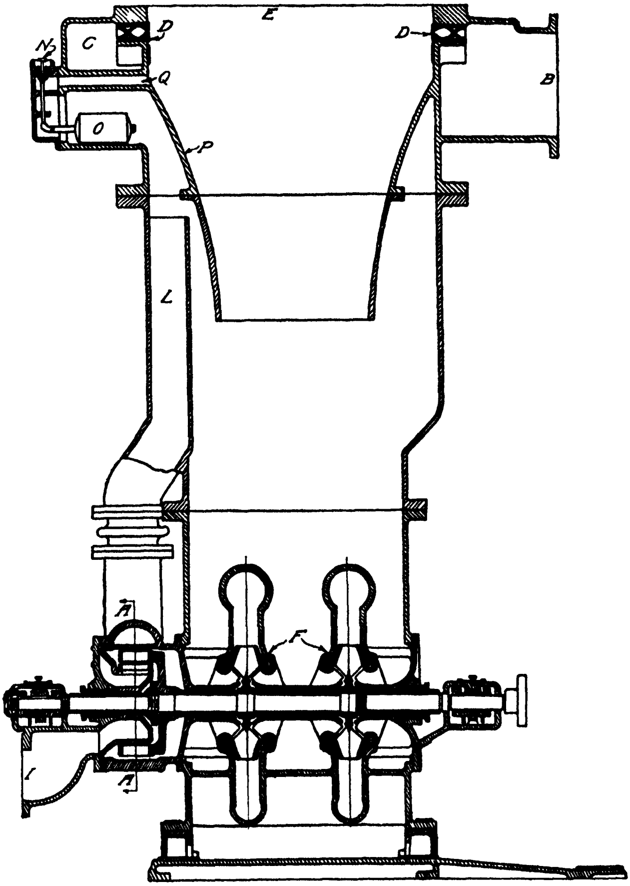


Fig 81 Cross Section of Leblanc Jet Condenser
Courtesy of Westinghouse Machine Company, East Pittsburgh, Pennsylvania

is intimately mixed with the steam, condenses it, and falls to the bottom of the condenser. From here the water pumps *F* discharge the water from the condenser. The air released in the condenser by the water and condensed steam rises underneath the cone *P*, and is drawn off through the pipe *L* by the air pump *A A*. The inlet

I is the separate water supply for this air pump, shown more in detail by Fig. 82, which is a cross section on *A A*.

In Fig. 82, the water entering through the center of the pump is discharged through the orifice *J* into a tapering pipe, the water being emitted in a succession of layers, as indicated at *G*. These layers of water are sometimes spoken of as being water pistons. The air coming through pipe *L* is caught between these layers of water and carried to the atmosphere through the long diffuser pipe *K*.

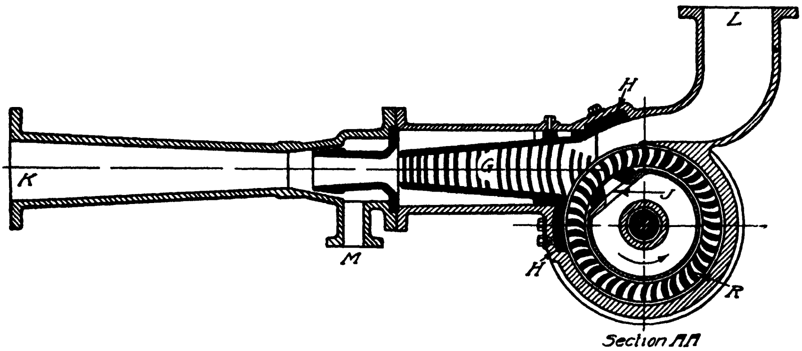


Fig 82 Section of Leblanc Condenser Taken through *AA*, Fig 81
Courtesy of Westinghouse Machine Company, East Pittsburgh, Pennsylvania

Since the cooling water enters by virtue of the vacuum, an accidental stopping of the pumps might cause serious trouble, due to the water rising above the top of the condenser. To take care of such emergencies, a very simple form of vacuum breaker is provided. In case the water rises in the condenser to an undesirable height, the float *O*, Fig. 81, opens the valve *N* and admits air to enter through passage *Q* directly into the condensing zone. This immediately stops the inflow of water by breaking the vacuum, and prevents damage to the turbine.

Relative Merits of Jet and Surface Condensers. In the jet condenser the steam, as soon as condensed, becomes mixed with the cooling water, and if the latter should be unsuitable for boiler-feed because of scale-forming impurities, acids, salts, etc., the pure distilled water represented by the condensed steam is wasted, and if it were necessary to purchase other water for boiler-feeding, this might represent a considerable waste of money. On the other hand, if the

cooling water is suitable for boiler-feeding or if a fresh supply of good water is easily obtainable, the jet condenser, because of its simplicity and low cost, is unexcelled. Surface condensers are recommended where the cooling water is unfitted for boiler-feed and where no suitable and cheap supply of pure boiler-feed water is available. Condensed steam from a surface condenser makes the best boiler-feed water, being in fact pure distilled water entirely free from scale-forming matter and containing a considerable amount of heat, as compared with cold feed water. If the exhaust from reciprocating engines is to be condensed and used as boiler-feed water, a suitable oil separator should be interposed in the exhaust pipe between the engine and the condenser. Another advantage of the surface condenser as compared with the jet condenser is that there is no danger, in case of failure of vacuum pumps, of the circulating water backing up into the engine cylinder and wrecking the engine.

Effect of Condenser on Efficiency. It has already been stated that there is a gain in thermal efficiency by running an engine condensing, but it will be more clearly seen by considering a few figures. The thermal efficiency may be expressed by the previously mentioned formula

$$E = \frac{T_1 - T_2}{T_1}$$

This efficiency may be increased if T_1 can be made larger—which would happen if the boiler pressure were increased—or if T_2 can be made smaller, which would result from reducing the back pressure by condensing. If the boiler pressure is raised, both the numerator and denominator of the fraction will increase, and the value of the fraction will be but slightly greater. If, however, the back pressure is reduced, the numerator $T_1 - T_2$ will be larger, while the denominator T_1 will remain the same. It is apparent that this will cause a much greater increase in efficiency than raising the boiler pressure a like amount.

Suppose an engine is supplied with steam at 85.3 pounds (gauge) pressure and it exhausts at 3.3 pounds (gauge) pressure. The absolute temperature corresponding to 85.3+14.7, or 100 pounds pressure, is 327.86+459.5, or 787.36 degrees, and the absolute temperature corresponding to 3.3+14.7, or 18 pounds pressure, is 222.40+459.5, or

TABLE I

Increase in Efficiency by Use of Condenser for Various Engines

Type of Engine	Feed Water per Indicated Horsepower				Per Cent Gained by Condenser
	Non-Condensing		Condensing		
	Probable Limits Pounds	Assumed for Comparison Pounds	Probable Limits Pounds	Assumed for Comparison Pounds	
Simple High Speed	35 to 26	33	25 to 19	22	33
Simple Low Speed	32 to 24	29	24 to 18	20	31
Compound High Speed	30 to 22	26	24 to 16	20	23
Compound Low Speed	— —	24	20 to 12½	18	25
Triple Exp. High Speed	27 to 21	24	23 to 14	17	29
Triple Exp. Low Speed	— —	—	18 to 12	—	—

681.9 degrees. Then the thermal efficiency determined from the formula becomes

$$E=\frac{T_1-T_2}{T_1}=\frac{787.36-681.9}{787.36}$$
$$=.134, \text{ or } 13.4 \text{ per cent}$$

If the boiler pressure were raised to 140 pounds absolute, the efficiency would be

$$E=\frac{812.59-681.9}{812.59}$$
$$=.161, \text{ or } 16.1 \text{ per cent}$$

If instead of increasing the boiler pressure a condenser is used and the exhaust pressure reduced to 4 pounds (absolute), the efficiency becomes

$$E=\frac{787.36-612.5}{787.36}=.222, \text{ or } 22.2 \text{ per cent}$$

Thus it is seen that if the exhaust pressure is lowered 14 pounds absolute there will be a greater increase in efficiency than if the boiler pressure is raised 40 pounds.

The per cent of efficiency that is obtained by the use of a condenser is shown in Table I.

Cost of Cooling Water Determines Condenser Economy. While the above figures are very encouraging, yet conditions may arise where the per cent of gain may be materially lessened or entirely lost,

due to the cost of water. Condensing engines require from 20 to 30 pounds of cooling water to condense each pound of steam used, depending on the necessary temperature. Thus it can be seen that the quantity of cooling water is relatively very large, and if it is purchased from a water company, quite an item is added to the yearly expense account for the one item of water. If, however, some means could be provided whereby the circulating water as it issues from the condenser could be cooled and then used over again in the condenser, the non-condensing engine could be run condensing, thus taking advantage of all the benefits due to the use of reduced back pressure and heating of the feed water. This has been attempted by conducting the heated discharge water to a pond, where it is allowed to cool to a lower temperature before being used again. Another plan is to place in the yard or on the roof of the building large shallow pans, in which the water is cooled by being exposed to the atmosphere. These methods are unsatisfactory on account of the considerable area necessary and the slow action. In addition, they are uncertain, because they are dependent upon atmospheric conditions.

Cooling Tower and Water Table. A more efficient and at the same time more expensive process is to use a cooling tower or a water table. Fig. 83 illustrates the general arrangement of a cooling tower located upon the roof of a building. The discharge from the condenser is led, as shown by the arrows, to the top of the cooling tower, where it is cooled before being returned to the condenser. This cooling is effected by distributing the water, by a system of piping, to the upper edge of a series of mats or slats, over the surface of which the water flows in a thin film to a reservoir which is situated in the bottom of the cooling tower. The mats partially interrupt the flow and, by breaking up the water in small streams, cause new portions to be exposed to the cooling effect of the air currents. The water from the reservoir then flows downward through the suction pipe and is pumped by the circulating pump through the condenser. After passing through the condenser and absorbing heat from the exhaust steam, it rises through the discharge pipe and commences the circuit over again.

The tower may have several arrangements and be made of various materials. A satisfactory form is constructed of steel plates

within the tower, or a large number of mats of steel wire cloth galvanized after weaving. The tower may be supported upon a proper foundation or upon legs, instead of being situated on the top of a building, as the one shown in the illustration.

To assist in the cooling of the water, the air is often made to circulate rapidly by means of a fan, which forces the air into the

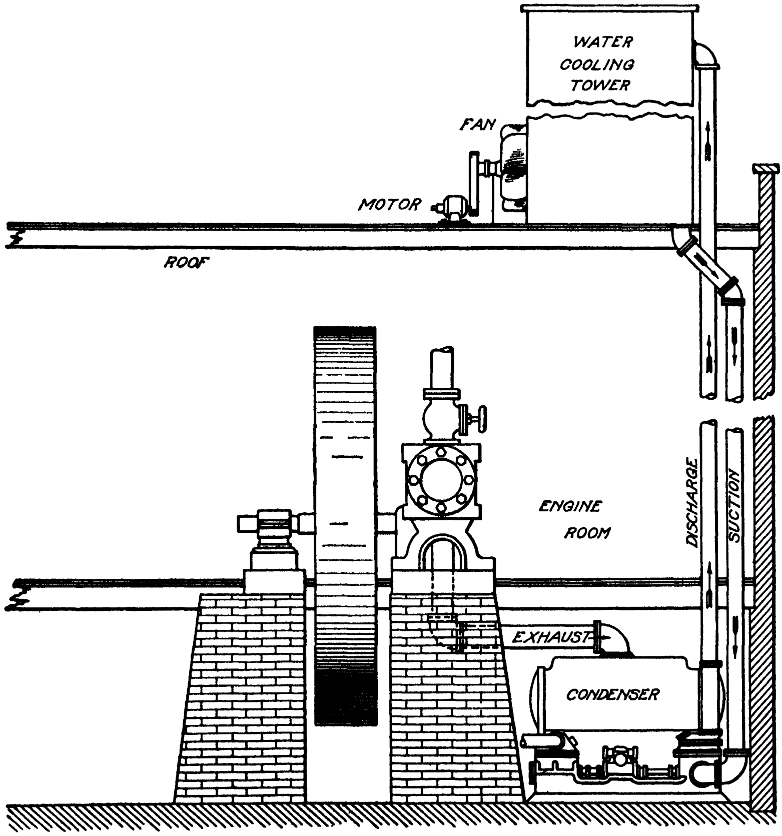


Fig 83. Diagram of Stationary Engine with Connections to Water Cooling Tower on Roof of Building

lower part of the tower and upward through the mats. This fan may be driven by an electric motor, by a line of shafting, or by a small independent engine.

In case the fan is not used, the mats are arranged so that they are exposed to the atmosphere. This of course necessitates the removal of the steel casing. Usually the fanless tower must be

placed at the top of a high building or in some position where the currents of air can readily circulate through the mats.

With an efficient type of cooling tower, the water may be reduced from 30 to 50 degrees, thus allowing a vacuum of from 22 to 26 inches. This will, of course, greatly increase the economy of the plant and allow the heated feed water to be returned to the boiler.

The water table is usually made of wooden slats placed in the ground near the plant. After trickling over the slats and becoming cooled by the air, it collects in the bottom of the reservoir and is then pumped into the condenser.

Amount of Cooling Water Per Pound of Steam. Besides condensing the steam, the injection water cools it still further, so that more than merely the latent heat is removed from it. If exhaust steam enters the condenser at a temperature t_1, it contains a certain amount of heat, known as *total heat at temperature* t_1. If it is condensed and cooled to a temperature t_2, at which it leaves the condenser, it then contains a certain amount of heat, known as *total heat at temperature* t_2.

If A represents the total heat at t_1 and B represents the heat of the liquid at t_2, then the heat given up by one pound of condensed steam is equal to $(A-B)$ British Thermal Units, provided the exhaust that enters the condenser is dry saturated steam. If C is the temperature of the injection or cooling water and D is the temperature of the discharge water, then every pound of cooling water absorbs approximately one British Thermal Unit for every degree rise in the temperature, or we may say that the heat absorbed is equal to $(D-C)$ British Thermal Units per pound of cooling water. Then it will take as many pounds of water W to absorb $(A-B)$ heat units as $(D-C)$ is contained in $(A-B)$. This may be expressed thus

$$W=\frac{(A-B)}{(D-C)}$$

Therefore, W represents the number of pounds of water required per pound of steam condensed.

EXAMPLE 1. Suppose steam is expanded in an engine to 4 pounds absolute pressure. If the initial temperature of the cooling water is 45 degrees, and the condenser is of the surface type, discharging water at 120 degrees,

and the temperature of the condensed steam is 130 degrees, how many pounds of cooling water are required per pound of steam?

SOLUTION. By consulting the steam tables, we find the total heat of steam at 4 pounds pressure to be 1126 5 British Thermal Units The heat of the liquid in the condensed steam at 130 degrees is 98.0 British Thermal Units. Then

$$W=\frac{1126\ 5-98\ 0}{120-45}$$
$$=13\ 71 \text{ pounds}$$

EXAMPLE 2. Suppose steam at 6 pounds absolute pressure exhausts into a jet condenser. The temperature of the injection water is 50 degrees and the discharge is 120 degrees. How many pounds of water are necessary to condense 8 pounds of steam?

SOLUTION. In the jet condenser the temperature of the condensed steam and the discharge water is the same. We find from the steam tables that the total heat of steam at 6 pounds absolute is 1133 6 British Thermal Units, and the heat of the liquid in the condensed steam at 120 degrees is 88 0 British Thermal Units. Then as before.

$$W=\frac{1133\ 6-88\ 0}{120-50}$$
$$=14\ 94$$

Therefore, 8 pounds of water will require $14\ 94\times8$, or 119.52 pounds.

The above calculation can not be relied upon to any great extent for we seldom know the true condition in the condenser, and it would be of little value to us if we did know, as the exact condition will change considerably. In practice it is customary to allow for about twice as much water as the above calculation would require. These figures give us a fair idea of the necessary sizes of the pipes and passages leading to the condenser, and give a basis for estimating the dimensions of the air pump.

Cooling Surface in Surface Condensers. The amount of surface required to condense the steam in surface condensers depends upon the conductivity of the metal, the condition of the tubes and their thickness, and the difference in temperature between the two sides. The tubes of a condenser are much thinner than boiler tubes, hence we might expect them to be more efficient in condensing the steam than the boiler tubes are in evaporating water. It has been found in actual practice, that a surface condenser receiving cooling water at 60 degrees and discharging it at 120 degrees will condense from 10 to 20 pounds of steam per square foot of the tube surface per hour. An average of 13 pounds per square foot of surface per hour

is considered a fair one. With exhaust pressure from 6 to 30 pounds absolute, it has been found that an allowance of 1.5 to 3.0 square feet of cooling surface per indicated horsepower is sufficient, when the initial temperature of cooling water is 60 degrees and the final temperature is 120 degrees.

It is evident that the amount of surface will depend upon the quantity of steam used per hour by the engine, the pressure and temperature of the exhaust, and the temperature of the cooling water and discharge. There must also be an allowance for inefficient work after the condenser has become fouled with service. All these conditions make the problem so uncertain that calculations by means of formulas are likely to be untrustworthy, and it is best at all times to make estimates from the figures given for similar conditions in actual service.

Feed Water Heaters. In many places where water is expensive and the condensing engines can not be run economically, a very considerable saving can be effected either by allowing the exhaust steam to condense into a feed water heater, thus saving the heat that would otherwise be wasted, or by using the exhaust steam for heating purposes. Of course in such cases the steam consumption of the engine is high, but if proper allowance is made for the heat used for other purposes, the actual fuel consumption rightfully charged to the engine is not excessive. If the feed water is heated by waste gases, then the gain belongs to the boiler and not to the engine.

ANALYSIS OF ENGINE MECHANISMS

CRANK EFFORT

In the steam engine the steam exerts a pressure on the crank pin through the piston rod and connecting rod. When the crank is at the dead center, the entire pressure is on the bearing of the crank shaft, and there is no tendency to turn the crank. As the crank pin moves from the dead center, the tendency increases until it reaches a maximum and then decreases until, at the other dead center, it is zero again. If the connecting rod were of infinite length and steam were admitted throughout the whole stroke, the maximum tendency, or the maximum turning moment as it is called, would occur with the crank at right angles to the line connecting the dead points.

Variable Thrust. In the actual engine the thrust along the rod is constantly varying even though the pressure on the piston remains the same. This is due to the angularity of the connecting rod. The turning moment is always equal to the thrust along the connecting rod multiplied by the perpendicular distance from the connecting rod to the center of the shaft. If the steam pressure on the piston remains constant, the maximum turning moment occurs when the connecting rod is at right angles to the crank, for in this position the perpendicular distance from the rod to the center of the shaft is a maximum and equal to the length of the crank; and, as the rod makes its greatest angle with the line connecting the dead center at this point, the thrust along it will also be a maximum. If the cut-off is very early, one-quarter stroke for instance, the maximum thrust along the rod will occur earlier than at the point previously mentioned, but the leverage of the force will be less, so that really there will be little change in the point of maximum turning moment no matter where the cut-off may occur.

Diagrams. To represent this turning moment, diagrams of crank effort may be drawn, with rectangular co-ordinates, having the crank angles represented as abscissas and the turning moments corresponding to these angles as ordinates.

FLYWHEEL

Besides the thrust of the connecting rod there must be taken into account friction and the inertia of the reciprocating parts. At first this may be thought of small consequence but with a fairly heavy piston and connecting rod it is obvious that at high speed the momentum would be great. In the case of a vertical engine, on the up stroke the steam must lift this heavy mass and impart a very considerable velocity to it, while on the down stroke the acceleration of the mass is added to the steam pressure. This makes the effective force on the up stroke less than that due to the actual steam pressure, and greater on the down stroke.

Function. In the case of a horizontal engine it is evident that while the piston can push the crank around during part of the stroke, and pull it along during another part, yet at the end of the stroke the pressure on the piston, no matter how great, can exert no turning moment on the shaft. Therefore, if some means is not pro-

vided for making the shaft turn past these points without the assistance of the piston, it may stop. This means is provided in the flywheel which is merely a heavy wheel placed on the main shaft. On account of the momentum of the flywheel it can not be stopped quickly and therefore carries the shaft around until the piston can again either push or pull.

Size of Wheel. If a long period be considered, the mean effort and the mean resistance must be equal; but during this period there are temporary changes of effort, the excesses causing increase of speed. To moderate these fluctuations several methods are employed.

The turning moment on the shaft of a single cylinder engine varies, *first*, because of the change in steam pressure, and *second*, on account of the angularity of the connecting rod. Before the piston reaches mid-stroke the turning moment is a maximum, as shown by

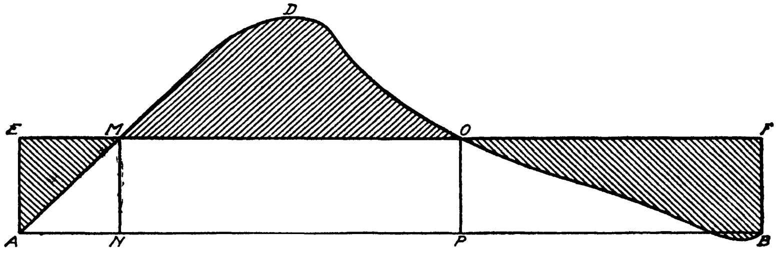


Fig 84 Graphical Representation of Turning Moment of Crank Shaft of a Single-Cylinder Engine for One Stroke

the curve, Fig. 84. Near the ends of the stroke the turning moment diminishes and finally becomes zero. This, of course, tends to cause a corresponding change in the speed of rotation of the shaft. In order to have this speed as nearly constant as possible and to give a greater uniformity of driving power, the engine may be run at high speed. By this means the inertia of the revolving parts, such as the connecting rod and crank, causes less variation. When the work to be done is steady and always in the same direction, a heavy flywheel may be used. The heavier the flywheel, the steadier will be the motion. It is desirable, of course, in all engines to have steady motion, but in some cases it is more important than in others. For instance, in electric lighting plants it is necessary that the machinery shall move with almost perfect steadiness. It is undesirable to use larger wheels than are absolutely necessary, because of the cost of the metal, the weight on the bearings, and the danger from bursting.

Methods of Reducing Size. If the turning moment which is exerted on the shaft from the piston could be made more regular and if dead points could be avoided, it would be possible to get a steadier motion with a much smaller flywheel.

If the engine must be stopped and reversed frequently, two or more cylinders are used, being connected to the same shaft. The cranks are placed at such angles that when one is exerting its minimum rotative effort, the other is exerting its maximum, or when one is at a dead center, the other is exerting its greatest effort. These cylinders may be identically the same in dimension as is the case with most hoisting engines and with many locomotives; or the engine may be compound or triple expansion. This arrangement is also used on engines for mines, collieries, and for hoisting of any sort where ease of stopping, starting, and reversing are prerequisites. Simple expansion engines with their cranks at right angles are usually spoken of as being coupled.

The governor adjusts the power of the engine to any large variation of the resistance. The flywheel has a duty to perform which is similar to that of the governor. It is designed to adjust the effort of the engine to sudden changes of the load which may occur during a single stroke. It also equalizes the variation in rotative effort on the crank pin. The flywheel absorbs energy while the turning moment is in excess of the resistance, and restores it while the crank is at or near the dead points. During these periods the resistance is in excess of the power.

Action of Flywheel. The action of the flywheel may be represented as in Figs. 84 and 85. It will be noticed that in Fig. 84, the curve of the crank effort runs below the axis toward the end of the stroke. This is because the compression is greater than the pressure near the end of expansion, and produces a resultant pressure on the piston. In Fig. 85 the effect of compression has been neglected. Let us suppose that the resistance, or load, is uniform. In Fig. 84, the line *A B* is the length of the semi-circumference of the crank pin. or the circumferential distance the crank pin moves during one stroke. The curve *A M D O B* is the curve of turning moment for one stroke. *M N* is the mean ordinate and, therefore, *A E F B* represents the constant resistance. The effort and resistance must be equal if the speed is uniform; hence the area *A E F B* equals

A M D O B. Then area *A E M* plus area *O F B* equals area *M D O*. At *A* the rotative effort is zero because the crank pin is at the dead point and from *A* to *N*, the turning moment is less than the resistance. At *N* the resistance and the effort are equal. From *N* to *P* the effort is in excess of the resistance. At *P* the effort and the resistance are again equal. From *P* to *B* the resistance is greater than the effort. In other words, from *A* to *N* the work done by the steam is less than the resistance. This shows that the work represented by the area *A E M* must have been done by the moving parts of the engine. From *N* to *P* the work done by the steam is greater than the resistance, and the excess of energy is absorbed by or stored in the moving parts. From *P* to the end of the stroke the work represented by the area *O F B* is done on the crank pin by the moving parts.

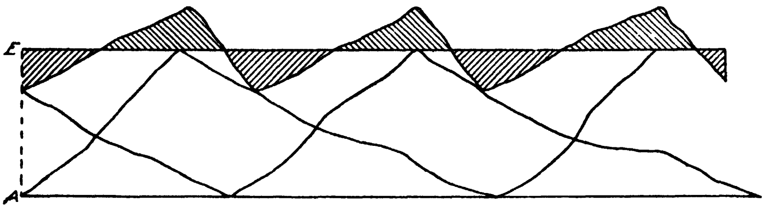


Fig 85 Simultaneous Crank Effort Curves of Two Engines Acting at Right Angles to Each Other

It is known that energy is proportional to the square of the velocity from the formula

$$E = \frac{WV^2}{2g}$$

in which E is energy in foot pounds, W is weight in pounds, V is velocity in feet per second, and g is acceleration of gravity in feet per second^2. Hence as W and g remain the same, the velocity must be reduced when the moving parts are giving out energy and increased when receiving energy. Thus it is seen that the action of the crank pin is to move slowly, then more rapidly. The weight of the revolving parts of an engine is not sufficient to absorb sufficient surplus energy, hence a heavy flywheel is used.

In case there are two engines at right angles, two crank effort curves may be drawn, as shown in Fig. 85. The mean ordinate *A E* is equal to the mean or constant resistance. There are two minimum and two maximum velocities in one stroke. The diagram shows

that the variation is much less than for a single cylinder, hence a lighter wheel may be used.

Calculations of Mass. The weight of the flywheel depends upon the character of the work done. For pumping engines and ordinary machine work the effort need not be as constant as for electric lighting. In determining the proper weight of a flywheel the diameter of the wheel must be known. If the wheel is too large, the high linear velocity of the rim will cause too great a centrifugal force and the wheel will not be safe. In practice, about 6,000 feet per minute is taken as the maximum linear velocity of cast-iron wheels. When made of wood and carefully put together the velocity may be taken as 7,000 to 7,500 feet per minute.

The linear velocity of a wheel is expressed in feet per minute by the formula $V=2\pi\ R\ N$, or $\pi\ D\ N$, in which V is velocity in feet per second, R is radius of wheel in feet, D is diameter of wheel in feet, and N is revolutions per minute.

Then if a wheel runs at 100 revolutions per minute, the allowable diameter would be obtained from the equation

$$6000=3.1416\times D\times 100$$

Therefore

$$D=\frac{6000}{3.1416\times 100}$$
$$=19.1 \text{ feet}$$

If a wheel is 12 feet in diameter the allowable speed is found to be

$$N=\frac{V}{\pi D}$$
$$=\frac{6000}{3.1416\times 12}$$
$$=159 \text{ revolutions per minute}$$

It is usual to make the diameter less than the calculated diameter.

Having determined the diameter, the weight may be calculated by several methods. There are many formulas to obtain this result given by various authorities, one formula being

$$W=\frac{C\times d^2\times b}{D^2\times N^2}$$

in which W is weight of rim in pounds; d is diameter of cylinder in inches; b is length of stroke in inches; D is diameter of flywheel in

feet; N is number of revolutions per minute; and C is a constant having a value which varies for different types of engines and for different conditions as follows:

Slide valve engines, ordinary work	$C = 350{,}000$
Corliss engines, ordinary work	$C = 700{,}000$
Slide-valve engines, electric lighting	$C = 700{,}000$
Automatic high speed engines	$C = 1{,}000{,}000$
Corliss engines, electric lighting	$C = 1{,}000{,}000$

EXAMPLE 1. Find the weight of a flywheel rim for an automatic high speed engine used for electric lighting The cylinder is 24 inches in diameter; the stroke is 2 feet. It runs at 300 revolutions per minute, and the flywheel is to be 6 feet in diameter.

SOLUTION.

$$W = \frac{1000000 \times (24)^2 \times 24}{36 \times 90000}$$
$$= 4266 \text{ pounds}$$

EXAMPLE 2 A plain slide valve engine for electric lighting is 20 inches × 24 inches It runs at 150 revolutions per minute. The flywheel is to be 8 feet in diameter What is the weight of its rim?

SOLUTION

$$W = \frac{700000 \times 400 \times 24}{64 \times 22500}$$
$$= 4666 \text{ pounds}$$

The weight of a flywheel is considered as being in the rim. The weight of the hub and arms is simply extra weight. Then, if the weight of the rim and its diameter be known, the width of the face and thickness of the rim can be found. Assume the given diameter to be the mean of the diameter of the inside and outside of the rim. Let b equal width of face in inches; t equal thickness of rim in inches; d equal diameter of flywheel in inches; and .2607 equal weight of 1 cubic inch of cast iron. Then

$$W = .2607 \times b \times t \times \pi d$$
$$= b \times t \times .819\, d$$

EXAMPLE 3. Suppose the rim of a flywheel weighs 6,000 pounds, is 9 feet in diameter, and the width of the face is 24 inches What is the thickness of the rim?

SOLUTION

$$t = \frac{W}{.819\, db}$$
$$= \frac{6000}{.819 \times 108 \times 24}$$
$$= 2.83 \text{ inches}$$

In this case the rim would probably be made $2\frac{13}{16}$ inches thick The total weight, including hub and arms, would probably be about 8,000 pounds.

GOVERNOR

The load on an engine is never constant, although there are cases where it is nearly uniform. While the engine is running at constant speed, the resistance at the flywheel rim is equal to the work done by the steam, disregarding friction. If the load on the engine is wholly or partially removed and the supply of steam continues undiminished, the force exerted by the steam will be in excess of the resistance. Work is equal to force multiplied by distance; hence, with constant effort, if the resistance is diminished, the distance must be increased. In other words, the speed of the engine will be increased, and the engine will "race." Also, if the load increases and the steam supply remains constant, the engine will "slow down."

It is evident, then, that if the speed is to be kept constant some means must be provided so that the steam supply shall at all times be exactly proportional to the load. This is accomplished by means of a governor.

Methods of Action. Steam-engine governors act in one of two ways (1) they may regulate the pressure of steam admitted to the steam chest, or (2) they may adjust the speed by altering the amount of steam admitted. Those which act in the first way are called *throttling governors,* because they throttle the steam in the main steam pipe. Those of the latter class are called *automatic cut-off governors,* since they automatically regulate the point of cut-off.

Theoretically, the method of governing by throttling the steam causes a loss in efficiency, but the throttling superheats the steam, thus reducing cylinder condensation. By the second method the loss in efficiency is very slight, unless the ratio of expansion is already great, in which case shortening the cut-off causes an increasing cylinder condensation.

Control by Centrifugal Force. In most governors of the throttling type and those applied to Corliss engines, centrifugal force counteracted by some other force is employed. A pair of heavy masses (usually iron balls or weights) are made to revolve about a spindle, which is driven by the engine. When the speed increases, the centrifugal force increases and the balls tend to fly outward, that is, they revolve in a larger circle. The controlling force, which is usually gravity or springs, is no longer able to keep the balls in

their former path. When, therefore, the increase is sufficiently great, the balls in moving outward act on the regulator, which may throttle the steam or cause cut-off to occur earlier.

With the throttling governor, a balanced throttle valve is placed in the main steam pipe leading to the valve chest. If the engine runs faster than the desired speed, the balls are forced to revolve at a higher speed. The increase in centrifugal force will cause them to revolve in a larger circle and in a higher plane. By means of levers and gears, the spindle may be forced downward, thus partially closing the valve. The engine, therefore, takes the steam at a low pressure, and consequently the speed falls slightly.

Similarly, if the load is increased, the engine slows down, causing the balls to drop and open the valve more widely; steam at higher pressure is then admitted and the speed is increased to the regular number of revolutions.

With the Corliss or other four-valve engines, the governor acts differently. Instead of throttling the steam in the steam pipe, the governor is connected to the releasing gear by rods. An increase of speed causes the releasing gear to unhook the disengaging link earlier in the stroke. This causes earlier cut-off, which of course decreases the power and speed, since the amount of steam admitted is less. If for any reason the load increases, the governor causes the valves to be held open longer. The cut-off, therefore, occurs later in the stroke.

Pendulum Governor. One of the most common forms of governor is similar to that invented by James Watt. It is called from its appearance the pendulum governor and is illustrated in principle in Fig. 86. To consider the theory of the pendulum governor, the masses of the balls are assumed to be concentrated at their centers and the rods are made of some material having no weight.

When the governor is revolving about its axis at a constant speed, the balls revolve in a circle having a radius r. The distance from this plane to the intersection of the rods, or the rods produced, is called the height and is equal to h.

If the balls revolve faster, the centrifugal force increases, r becomes greater, and h diminishes. The mathematical expression for centrifugal force is

$$F = \frac{Wv^2}{gr}$$

in which F is force in pounds; W is weight of one ball in pounds; v is velocity in feet per second; g is acceleration due to gravity; and r is radius in feet. From the above equation it is seen that force varies inversely as the radius.

While the pendulum is revolving, centrifugal force acts horizontally outward and tends to make the balls fly from the center; and the action of gravity tends to make the balls drop downward. In order that the balls shall revolve at a certain height, the moments of these two forces about the point of suspension must be equal, or

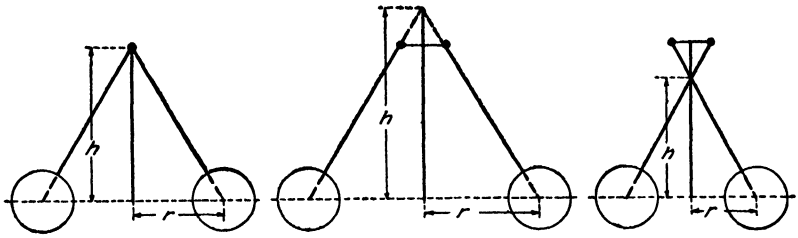


Fig. 86. Diagrams Showing Action of Pendulum Governor

the weight of the balls multiplied by their distance from the center must equal the centrifugal force multiplied by the height, or

$$W \times r = F \times h$$

from which

$$\frac{h}{r} = \frac{W}{F}$$

Substituting value of F just given, we have

$$\frac{h}{r} = \frac{W}{\frac{Wv^2}{gr}}$$

$$= \frac{gr}{v^2}$$

Therefore,

$$h = \frac{gr^2}{v^2}$$

Now since v, the linear velocity of a point revolving in the circumference of a circle, is expressed as $2\pi r N'$ feet per second, where N'

is revolutions per second, this value may be substituted in the above formula, giving

$$h=\frac{g\,r_2}{4\,\pi^2 r^2 (N')^2}$$

$$=\frac{g}{4\,\pi^2\,(N')^2}$$

and since the values of g and π are known, the formula may be written

$$h=\frac{32.16}{4\times 3.1416^2\times (N')^2}$$

$$=\frac{.8146}{(N')^2}\text{ feet}$$

$$=\frac{9.775}{(N')^2}\text{ inches}$$

If it is desired to use N, the r.p.m., instead of N', the r.p.s., the former may be substituted in the formula by multiplying the fraction by $\overline{60}^2$, or 3600, giving

$$h=\frac{2932.56}{N^2}\text{ feet}$$

$$=\frac{35190.7}{N^2}\text{ inches}$$

From the above formula it is evident that the height is independent of the weight of the balls or the length of the rod, depending entirely upon the number of revolutions. The height varies inversely as the square of the number of revolutions.

The ordinary pendulum governor is not isochronous, that is, it does not revolve at a uniform speed in all positions, the speed changing as the angle between the arms and spindle changes.

Fly-Ball Governor. The early form consisted of two heavy balls suspended by links from a pin connection in a vertical spindle, as shown in Figs. 87 and 88. The spindle is caused to revolve by belting or gearing from the main shaft, so that as the speed increases, centrifugal force causes the balls to revolve in a circle of larger and

larger diameter. The change of position of these balls can be made to affect the controlling valves so that the admission or throttling will vary with their position. With this governor it is evident that for a given speed of the engine there is but one possible position for the governor, consequently one definite amount of throttling or one point of cut-off, as the case may be. If the load varies, the speed of the engine will change. This causes the position of the governor balls to be changed slightly, thus altering the pressure. But in order that the pressure or cut-off shall remain changed, the governor balls must stay in their new position. That is to say, the speed of the engine

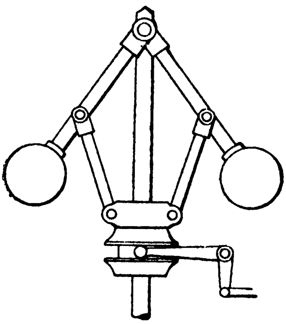


Fig 87. Simple Type of Fly-Ball Governor

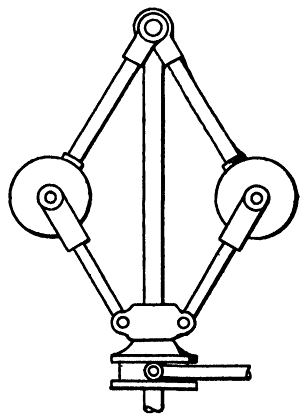


Fig 88 Later Type of Fly-Ball Governor

must be slightly changed. Thus with the old ball governors there was a slightly different speed for each load. This condition has been greatly improved by various modifications until now such governors give excellent regulation.

While the engine is running with a light load, the valve controlled by the governor will be open just enough to admit steam at a pressure that will keep the engine running at a given speed. Now if the engine is heavily loaded, the throttle valve must be wide open. The change of opening is obtained by a variation in the height of the governor, which is caused by a change of speed. Thus it is seen that the governor can control the speed only within certain limits which are not far apart. The difference in the extreme heights of the governor must be sufficient to open the throttle its entire range. In

TABLE II

Heights of Governor for Different Speeds of Engine

Number of Revolutions Per Minute	Height in Inches	Variation of Height in Inches 4 Per Cent
250	563	0225
200	879	035
175	1 149	046
150	1 564	062
125	2.252	090
100	3.519	140
75	6 256	250
50	14 076	563

most well-designed engines, equipped with a throttling governor, the speed will not vary more than 4 per cent, that is, 2 per cent above or below the mean speed.

From the formula $h=\frac{35190\ 7}{N^2}$, the heights corresponding to given speeds can be computed as shown in the second column of Table II. The third column is the variation in height for a speed variation of 4 per cent or 2 per cent either above or below the mean.

Disadvantage of Ordinary Fly-Ball Type. From Table II it will be seen that for a considerable variation of speed there is but slight variation in the height of the governor, this being too small to control the cut-off or throttling mechanism throughout the entire range. Also for high speeds the height of the governor is so small that it would be difficult to construct it.

Other disadvantages of the fly-ball governor are as follows: It is apparent that the valves must be controlled by the weight of the governor balls. In large engines this requires very heavy balls in order to quickly overcome the resistance of the valves. But these large balls have considerable inertia and will therefore be reluctant to change their speed with that of the engine. The increased weight will also increase the friction in the governor joints and the cramping action existing when the balls are driven by the spindle will increase this friction much further. All these things tend to delay the action of the governor, so that in all large engines the old-fash-

ioned governor became sluggish. The balls had to turn slowly because they were so heavy; this was especially troublesome in high-speed engines.

Porter Improved Type. To remedy these defects the weighted or Porter governor, Fig. 89. was designed. It has a greater height for a given speed, and the variation in height for a given variation of speed is greater and, consequently, more sensitive. By increasing this variation in height, the sensitiveness is increased. Thus, if a governor running at 50 revolutions has a variation in height of .57 inch, it is not as sensitive as one having a variation of 1 inch for the same speed.

In the weighted governor, the weight is formed so that the center of gravity is in the axis. It is placed on the spindle and is free to revolve. The weight adds to the weight of the balls, and thus increases the moment of the weight. It does not, however, add to the centrifugal force, and hence the moment of this force is unchanged. It may then be said that the weight adds effect to the weight of the governor balls but not to the centrifugal force, and as a consequence the height of the governor for a given speed is increased. If W equals the weight of the ball as before, and W' equals one-half the added weight, the equated moments are

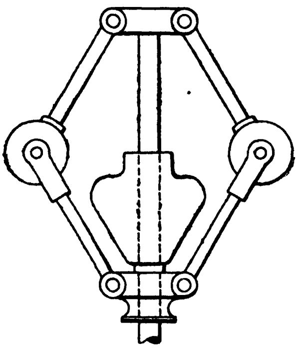


Fig 89. Porter Improved Type of Fly-Ball Governor

$$(W+W')\ r = Fh$$

Substituting for F its value obtained from the formula, p. 147, we have

$$(W+W')\ r = \left(\frac{Wv^2}{gr}\right) h$$

$$h = (W+W')r\left(\frac{gr}{Wv^2}\right)$$

$$= \frac{(W+W')r^2g}{W \times 4\pi^2 r^2 (N')^2}$$

$$= \frac{(W+W')}{W} \times \left(\frac{g}{(4\pi^2 (N')^2}\right)$$

Since it is known that

$$\frac{g}{4\pi^2(N')^2}=\frac{.8146}{(N')^2}$$

$$h=\left(\frac{W+W'}{W}\right)\times\frac{.8146}{(N')^2}$$

Hence the height of a weighted governor is equal to the height of a simple pendulum governor multiplied by $\left(\frac{W+W'}{W}\right)$, or $\left(1+\frac{W'}{W}\right)$.

For instance, if the height of a simple pendulum is 10 inches and

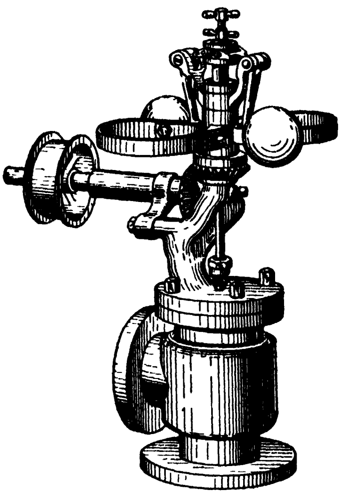


Fig 90. Waters Governor with Safety Stop

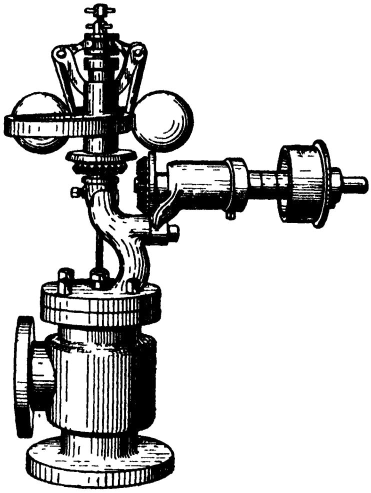


Fig 91. Waters Spring Type of Fly-Ball Governor

the weight of the balls equal to the added weight, the height of the weighted governor will be

$$h=\left(1+\frac{1}{1}\right)\times 10$$

$$=2\times 10$$

$$=20$$

Thus it is evident that if a weight equal to the combined weight of the balls is added, the height of the governor will be doubled. If the belt driving the governor slips off or breaks, the balls will

drop, with the result that the engine will "run away." To diminish this danger many governors are provided with some kind of safety stop which closes the valve when the governor loses its normal action. Usually a trip is provided which the governor does not touch in its normal positions, but which will be released if the balls drop down below a certain point.

Spring Type. In many cases a spring is used in place of the weight. This type of governor is frequently used on throttling engines, and it consists of a pendulum governor with springs added to counteract the centrifugal force of the balls. Thus the height and sensitiveness are increased. Fig. 90 shows the exterior view of a Waters governor and Fig. 91 shows the same governor having the safety stop. In this governor the weights are always in the same plane, the variation in height being due to the action of the bell-crank levers connecting the balls and spindle. When the balls move outward, the spindle moves downward and tends to close the valve. The governor balls are caused to revolve by means of a belt and bevel gears. The valve and seat are shown in section in Fig. 92. The valve is a hollow cylinder with three ports through which steam enters. The seat is made in four parts, that is, there are four edges that the steam passes as it enters the valve. The valve, being cylindrical and having steam on both sides, is balanced, and because of the many openings only a small travel is necessary.

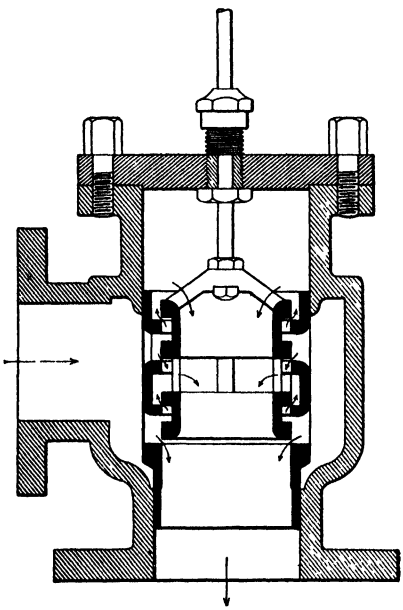


Fig. 92 Section of Valve and Valve Seat of Waters Governor

Shaft Governor. Usually some form of pendulum governor is used for throttling engines. For governing an engine by varying the point of cut-off, shaft governors are generally used, although the Corliss and some other engines use pendulum governors for this pur-

pose. Cut-off governors, which are called shaft governors because they are placed on the main shaft, are made in many forms, but their essential features are the same. Two pivoted masses or weights are arranged symmetrically on opposite sides of the shaft and their tendency to fly outward when the speed increases is resisted by springs. When in action the outward motion of the weights causes the admission valve to close earlier, and the inward motion causes it to close later. This change is effected by altering the position of

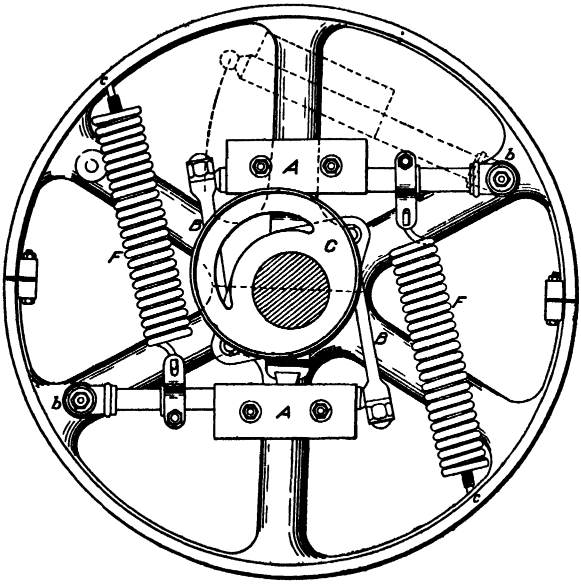


Fig 93 Diagram Showing Action of Buckeye Shaft Governor

the eccentric, either by changing the eccentricity or the angular advance.

Shaft governors are made in a great variety of ways, no two being exactly alike. If the principles of a few types are understood, it is easy to understand others.

Buckeye Type. The valve of the Buckeye engine is hollow and of the slide valve type. The cut-off valve is inside. The change of cut-off is due to the alteration of the angular advance, the arrangement of the parts which effect this alteration being shown in Fig. 93. A wheel which contains and supports the various parts of the gov-

ernor is keyed to the shaft. Two arms, having weights *A A* at the ends, are pivoted to the arms of the wheel *b b*. The ends having the weights are connected to the collar on the loose eccentric *C* by means of rods *B B*.

When the weights move to the position indicated by the dotted lines, the eccentric is turned on the shaft about a quarter of a revolution in the direction in which the engine runs, that is, the eccentric is advanced, or the angular advance is increased; this makes cut-off occur earlier, as shown by the table presented in "Valve Gears." If the engine had a single plain slide valve, the variation of the angular

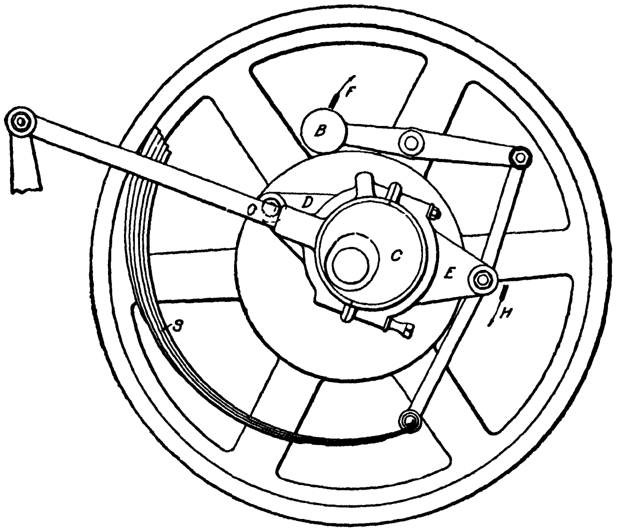


Fig 94 Diagram Showing Action of Straight-Line Type of Shaft Governor

advance would produce too great a variation of lead; but as this engine has a separate valve for cut-off, admission is not altered by the cut-off valve.

The springs *F F* balance the centrifugal force of the weights; the weights *AA* are varied to suit the speed; and the tension on the springs is altered by means of the screws *c c*. Auxiliary springs are added in order to obtain the exactness of regulation necessary for electric lighting. These springs tend to throw the arms outward, but act only during the inner half of this movement.

Straight-Line Typ?. Fig. 94 shows the governor of the Straight-line engine. It has but one ball *B*, which is linked to the spring *S*

and to the plate *DE*, on which is the eccentric *C*. When the ball flies outward in the direction indicated by the arrow *F*, the eccentric is shifted about the pivot *O*, the links moving in the direction of the arrow *H*. The ball is heavy and at a considerable distance from the center, hence it has a great centrifugal force and the spring must be stiff. The governor of the Buckeye engine alters the cut-off by changing the angular advance, while the Straight-line engine governor changes the travel of the valve. The latter type of valve is very common.

Inertia Form. The well-known Rites inertia governor, Fig. 95, is a form of shaft governor largely used for certain types of engines. This governor regulates the speed of the engine by shifting the eccentric, thus changing the valve travel and increasing or decreasing the angular advance, depending on the speed conditions. It differs in its operation from the centrifugal shaft governor previously considered, in that it makes use of the inertia of two large weights instead of centrifugal force. To understand this action, it first becomes necessary to know something about its construction.

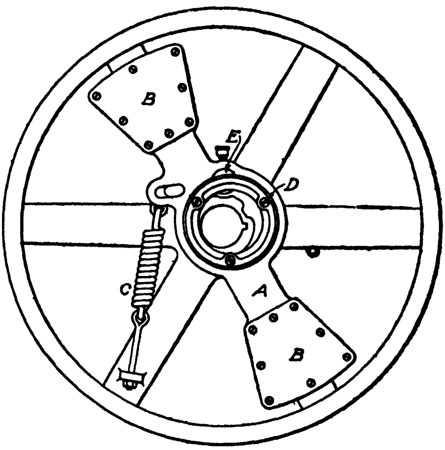


Fig 95. Diagram of Rites Inertia Type of Shaft Governor

The governor consists essentially of a heavy arm *A* pivoted at *E* to the flywheel. This arm carries two heavy weights at *B*. The eccentric *D* is fastened to the arm by three countersunk screws, as shown, and moves with reference to the engine shaft whenever the weights *B* cause the arm *A* to move about its pivot point. Fastened to the flywheel arm and the governor arm *A* is the spring *C*, which brings the arm *A* back to its normal position when the engine is not operating. This spring also has certain other functions to perform in the operation of the governor.

The action of the governor is such that the valve experiences very much the same movement as in the centrifugal governor. As

the engine speeds up, the tendency of the heavy arm A is to lag behind the flywheel. This lagging action controls the position of the eccentric so that the valve travel is reduced, thus limiting the amount of steam that enters the cylinders. If, after the engine is operating at a uniform rate of speed, an increase of load suddenly occurs, the motion of the engine shaft and flywheel will be slightly retarded and the engine will commence to "slow down." On account of the energy stored up in the governor arm and the weights BB, they will not be so quickly affected, hence the governor will be moving slightly faster than the shaft. As a result the eccentric position with reference to the shaft will be changed, and the valve travel increased, thus permitting more steam to enter the cylinder, increasing the power commensurate with the added load. If for any reason the engine takes a sudden spurt in speed, the tendency of the governor is to fall backward, so to speak; and if the engine is suddenly slowed down for any cause, the tendency of the governor is to plunge forward; hence the valve travel is shortened or lengthened according to which action takes place. This type of governor gives very close regulation when properly constructed.

ERECTION AND OPERATION OF STEAM ENGINES

The limited scope of this work will not permit of an exhaustive study of these two important details—the erection and operation of steam engines; only the general principles governing each will be pointed out.

ERECTION

Foundations. When about to erect an engine the first requisite is the foundation, the character of which will, of course, depend upon the type and the size of the engine. It should be built according to plans submitted by the engine builders, no changes of material consequence being made without the approval of the builders. It should be neither connected with nor in close proximity to any supporting column or columns of the building, as vibrations of the engine will be transmitted to the building which might prove to be disastrous. The foundation should be built upon a solid bottom, but if this is not obtainable at the depth required by the foundation plans, the base of the foundation should be extended in all directions in

order that the bearing surface may be increased. In the case of the horizontal engines the nearer the center of gravity of the foundation is placed to the center line of the engine, the more effective will be the foundation. In such cases, therefore, it is preferable to have an extended bearing surface rather than one of considerable depth. The foundation bolts and washers should be carefully located in accordance with the furnished plans. A space of one inch or more should be left around each of the foundation bolts. This may be obtained by using pieces of short iron pipe or old boiler tubes around the bolts, care being taken that they do not extend above the foundation, so as to prevent the proper tightening of the bolts after the engine is placed in position. After the engine is properly set, the space left around the foundation bolts should be filled with the best cement mortar, so as to insure their permanency. The foundation should be a solid one and built of brick, stone, or concrete.

Brick. When brick is used, a hollow square effect may be constructed and the open space filled with a mixture of concrete, consisting of one part cement and three parts sand and gravel.

Concrete. When making a concrete foundation, suitable forms must first be constructed to receive the concrete. Crushed stone or clean gravel or both may be used, care being taken to wash the gravel free of all clay. A good mixture for ordinary foundations is one having the proportions: $1:2\frac{1}{2}:5$. That is, 1 barrel, or 4 bags, cement, $2\frac{1}{2}$ barrels, or 9.5 cubic feet, of sand, and 5 barrels, or 19 cubic feet, of gravel or stone. If the foundation is to be waterproof, careful consideration must be given to the proportioning of the mixture. If the foundation covers considerable area and is not very deep, the mixture should be richer in cement; if, however, the foundation is very deep, a poorer mixture may be used at the bottom and a richer one near the top.

The cement, gravel or stone, and sand should first be thoroughly mixed in the dry state and the water added while the mixing process continues until the mass is well mixed and thoroughly wet. After the mixing is complete, the concrete should be laid in layers from 6 to 9 inches deep and well rammed until solid. The ramming of the concrete is an absolute necessity in order that a solid foundation may be secured.

When the foundation has been completed in accordance with the furnished plans, sufficient time must elapse before any machinery

is placed thereon in order to insure a proper setting of the cement. When the concrete has set sufficiently, it should be inspected to see that no omissions or errors have been made, after which the engine may be unpacked and prepared for setting. If the foundation is a large one, an inspector should be on hand at all times to follow the work and see that no errors are made.

Setting the Engine. Upon the accuracy and thoroughness of the setting of the engine, in a large measure depends its successful operation as to smoothness and efficiency of running. In this process there are a great many things to be considered. First, the base and sub-base must be carefully cleaned and set in position. Next, the crank shaft, cylinders, piston, crosshead valves, and other details must be carefully placed in position and alignment made according to the plans of the builders. As all of these details require skill, an inexperienced person should not attempt the setting up of an engine. It is always preferable, when possible, to obtain an experienced man from the engine builders.

Installation of Attachments. In addition to the erection and setting of the engine proper there are various attachments and auxiliaries that require care and skill in their proper installation. The steam and exhaust piping as well as the cylinder drainage should be carefully attended to. The piping should be of ample size, all bends should be easy, and gate valves should be used whenever possible. The piping should have a gradual fall from the boiler to the engine, at or near which should be placed a separator.

Separator. The separator should be of approved design, and care must be taken to carefully provide for drainage in order to insure the removal of the water, otherwise the separator might form a reservoir for water and thus endanger the engine more with its use than without. In addition to being a safeguard against water hammer, when properly attached, the separator also improves the steam economy of the engine, since it removes the most of the entrained moisture which is carried from the boiler through the steam pipes.

Exhaust Pipes. The exhaust pipes should be of ample area to take care of all exhaust steam, and safeguards should be used to insure no backing up of the condensed exhaust into the cylinders. To this end, sharp bends should be avoided and gate valves should be used

if valves are necessary, as by their use the area of the pipe is less reduced than by other forms of valves. Check valves should be avoided whenever possible.

Cylinder Drains. The cylinder drains should be of sufficient area to care for all condensed steam in the cylinders and so attached to the cylinders and the exhaust pipe or receiver that no pockets will be formed for the accumulation of water. In the case of compound engines the cylinder drains of the high and the low pressure cylinders should not be connected together, but separately connected to the exhaust or other main drain. In condensing engines the cylinder drains should always go into the exhaust drain if it is low, enough to admit of proper drainage.

OPERATION

Let us now turn our attention to the operation and management of an engine. It should be borne in mind that many suggestions as to the proper alignment and adjustment of bearings, the adjustment of valves, and the consideration given lubrication will be applicable both to the first setting up of the engine and also to the daily operation afterwards.

Competent Engineer a Requisite. The operation of an engine should be committed to a careful, skillful, and reliable man. This is especially true in the case of modern well-equipped plants which represent quite an outlay of capital. In many of the smaller plants, however, not much attention is given to the matter and we find, as a result, men holding positions as operators who know very little about their business. Under such conditions the plants are seldom operated efficiently.

As a suggestion of some of the duties of a man in charge of a modern plant, which also suggest the amount of judgment and experience required, the following general instructions are presented.

Care of Bearing Caps. The caps on the main bearings should always have sufficient liners underneath to enable the nuts on the bearing studs to draw the cap down tightly upon them and not pinch the shaft, which should be free to revolve in its bearings without unnecessary play.

The caps should be removed occasionally as conditions demand in order to clean out the oil grooves which are chipped in the babbitt

metal, as the passages may become clogged with dirt or other foreign matter.

Adjustment of Connecting Rod Box. In adjusting the connecting rod box at the crank pin end, the same general rules should be observed regarding the liners under the cap—the large nuts drawn solidly upon it, the small nuts firmly jammed and the cotter pins placed in position. The adjustment of the box should then be tested with a lever about 12 inches in length, the adjustment being so made that with a lever of this length the operator can easily move the end of the connecting rod sufficiently to take up the side play between the flanges on the crank pin and the end of the box. The adjustment should never be made so close that this side movement can not be observed.

The adjustment of the connecting rod box at the crosshead pin should be made by placing the crank on the center nearest the cylinder; then with a wrench provided for that purpose, slack off both wedge screws at the upper and lower sides of the connecting rod, and draw the wedge up until it is solid against the box; then slack off one screw about a sixth of a turn, and draw up the other so as to firmly lock the wedge.

Lining Up Crosshead. The crosshead should be lined up between the guides, while disconnected from the connecting rod. When in this condition the crosshead should be so lined that it can be easily pulled from one end of the guides to the other with a short lever.

The crosshead should never be run very close, and should always be free enough to allow long and continuous runs without heating the guides to the degree that they would be uncomfortably warm to the touch.

When making any adjustments of the crosshead, the operator should assure himself that the lock nut which prevents the piston rod turning in the boss of the crosshead is securely placed.

Adjusting Eccentric Strap. The eccentric strap adjustment is made by liners placed between the halves of the strap and double nutted bolts. When adjustment is necessary, the other end of the eccentric rod should be disconnected and, after drawing up the strap bolts, it should be tested by giving the strap a half revolution about the eccentric. If it is found that the friction between the strap and the eccentric is sufficient to support the weight of the rod, the bolts

should be loosened and liners replaced until the strap moves freely without lost motion. The double nuts should then be locked and the cotter pins replaced in the ends of the bolts.

Governor. The governor should be adjusted to meet the different conditions of speed and steam pressure and the degree of regulation required. As governors differ so much in design and detail of construction, it is not possible to give any general rule for their adjustment. The operator, if desired, can usually obtain instructions from the engine builder for the particular type of governor in question.

Valve Setting. As a discussion of the setting of the valves and their adjustment for wear will be found given in "Valve Gears," no consideration of the subject will be presented here.

Lubrication. The lubrication of a steam engine, and especially of high speed engines, is a very important consideration with both the designer and the operator, for it is upon proper lubrication that they must largely depend for a constant and satisfactory operation. The designer must, therefore, provide ample and efficient facilities for lubricating the bearings, cylinders, and valves, whereas the operator must use discretion in selecting his lubricants and the amount to use after selection has been made.

Choice of Oils. It might be said that only the best oils should be used. Cheap oils are usually considered expensive at any cost and should be avoided as they promote excessive wearing of the parts—causing noisy operation--and may cause serious cutting of the cylinders. There are two general classes of liquid lubricants now in the market, namely, mineral and animal oils. There is also a compounded lubricant which is made up of about 5 to 15 per cent of animal matter and the balance of mineral oil. This compound makes a very efficient lubricant for some classes of service, as it withstands the action of the condensation and adheres to the surface of the cylinders, thus giving better results than larger quantities of mineral oil.

In plants where open heaters are used and where the exhaust steam is condensed and used for boiler feed water, the compounded oils can not be used, on account of the danger of the animal matter getting into the boilers and causing considerable trouble. In such cases mineral oil must be used, although it may require considerable more mineral than compounded oil to accomplish the lubrication.

Solid Lubricants. Several solid lubricants are used, such as graphite, metalline, soapstone, and fiber graphite.

Graphite when mixed with certain oils is well adapted for heavy pressures. It is especially good for heavy pressures and low velocities. Under conditions which require a large amount of cylinder oil, a small amount of crystal or flake graphite may be used with good results. Care must be exercised, however, if the exhaust steam is used for feed water, as the graphite may get into the boilers and cause inconvenience and perhaps serious trouble.

Metalline is a solid compound containing graphite. It is made in the form of solid cylinders, which are fitted to the holes drilled into the surface of the bearing. When a bearing is thus fitted, no other lubricant is necessary.

Soapstone in the form of powder and mixed with oil or fat is sometimes used as a lubricant. Soap mixed with graphite or soap-stone is often used where wood is in contact with wood or iron.

A preparation called *fiber graphite* is used for self-lubricating bearings. It is made of finely divided graphite mixed with fibers of wood. It is pressed in molds and afterwards fitted to bearings.

For great pressure at slow speed, graphite, lard, tallow, and other solid lubricants are suitable. If the pressure is great and the speed high, castor, sperm, and heavy mineral oils are used.

For low pressure and high speed, olive, sperm, rape, and refined petroleum give very satisfactory results.

In ordinary machinery, heavy mineral and vegetable oils and lard oil are good. The relative value of various lubricants depends upon the prevailing conditions. Oil that is suitable for one place might not flow freely enough for another.

The quality of oil is of great importance. In many branches of industry it is imperative that the machinery run as perfectly as possible. On this account and because of the high cost of machinery, only first class oil should be used. The cylinder oil especially should be high grade, because the valves, piston, and piston rods are the most delicate parts of the engine.

Qualities of a Good Lubricant. From the foregoing brief discussion of lubricants it will be evident that they must possess certain qualities which may be enumerated as follows:

The lubricant must be sufficiently fluid, so that it will not in itself make the bearing run hard.

It must not be too fluid or it will be squeezed out from between the bearing surfaces. If this happens, the bearing will immediately heat and begin to cut. The heating will tighten the bearing and increase the pressure and the cutting.

It must not gum or dry when exposed to the air.

It must not be easily decomposed by the heat generated. If it should be decomposed, it might form substances which would be injurious to the bearings.

It must not take fire easily.

It must contain no acid and should form no acid in decomposing, as acids corrode the bearings.

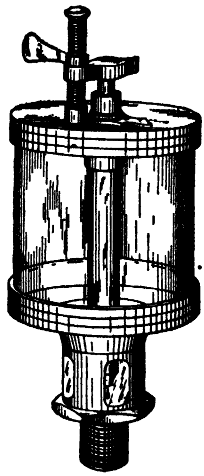


Fig 96 Standard Type of Simple Oil Cup

Both mineral and animal oils are used as lubricants. Formerly animal oils were used entirely, but they were likely to decompose at high temperatures and form acids. It is important in using high pressure steam to have "high test oils," that is, oils which will not decompose or volatilize at the temperature of the steam. It was the difficulty of getting such oils which made great trouble when superheated steam was first used. Mineral oils will stand high temperatures very readily, and even if they do decompose, they form no acids.

Common Oilers. Engines are lubricated by means of oil cups and wipers placed on the bearings wherever required. They are made in many forms. Formerly, the oil cup was made with a tube extending through the oil. A piece of lamp wick or worsted leads from the oil in the cup to the tube. Capillary attraction causes the oil to flow continuously and drip down the tube. When not in use, the lamp wick should be withdrawn. This type of oil cup is now seldom used.

The oil cup shown in Fig. 96 is simple and economical. The opening of the valve is regulated by an adjustable stop. The oil may be seen as it flows drop by drop. The cylindrical portion is made of glass, so that the operator can see how much oil there is in the cup without opening it.

A form of wiper crank pin oiler is shown in Fig. 97. The oil cup is attached to a bracket. The oil drops from the cup into the sheet of wicking or wire cloth and is removed at each revolution of the crank pin by means of the cup which is attached to the end of the connecting rod. This form of oiler works very satisfactorily at slow speeds.

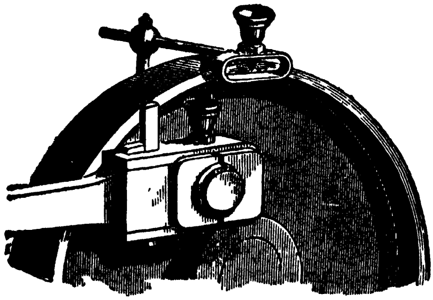


Fig 97. Form of Wiper Crank Pin Oiler

Centrifugal Oilers. Fig. 98 shows a centrifugal oiling device which operates very satisfactorily at all speeds. The oil flows from the oil cup through the tube to the small hole in the crank pin by centrifugal force. It reaches the bearing surface by means of another small hole.

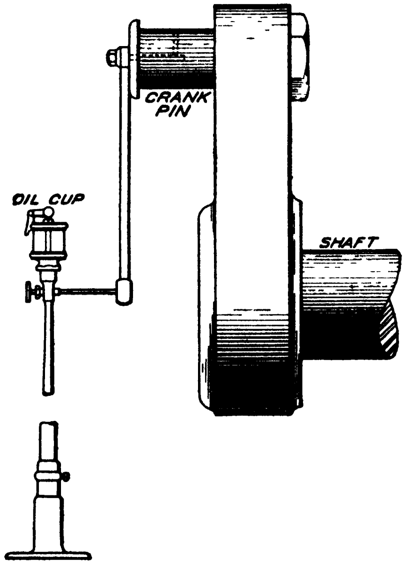


Fig 98. Centrifugal Oiler

Cylinder Lubrication. In oiling the valve chest and the cylinder, the lubricant must be introduced against the pressure of the steam. This may be done in several ways, in each of which it is introduced into the steam before it reaches the valve chest and is carried by the steam to the surfaces to be lubricated.

By Oil Pumps. The oil may be forced into the steam pipe by a small hand pump or, in large engines, by an attachment from the engine itself. The supply of oil is, of course, intermittent if the pump is driven by hand, but continuous and economical if driven by the engine.

By Sight-Feed Lubricators. The most common device for feeding oil to the cylinder is that which introduces the oil drop by drop into the steam when it is in the steam pipe or steam chest. The oil

becomes vaporized and lubricates all the internal surfaces of the engine.

Fig. 99 shows the section of a sight-feed lubricator, which must be placed on the steam supply pipe in a vertical position above the throttle. The reservoir *O* is filled with oil. The pipe *B*, which connects with the steam pipe, is partly filled with condensed steam which flows down the small curved pipe *E* to the bottom of the chamber *O*. A small portion of the oil is thus displaced and flows from the top of the reservoir *O* down the tube *F* by the regulating valve *D*, and up through the glass tube *S*, which is filled with water. It enters the main steam pipe through the connection *A*. The gauge glass *G* indicates the height of water in the chamber *O*. To fill the lubricator, close the regulating valve *D* and the valve in pipe *B*; the oil chamber can thus be drained through the cock *C*, and filled. If the glass *S* becomes clogged, it may be cleaned by closing valve *D* and opening the small valve *H*. This will allow the steam to blow through the glass. After cleaning close valve *H* and allow glass *S* to become filled with water before opening the feed valve. The amount of oil fed to the cylinder can be regulated by opening the valve *D* the proper amount. The exact quantity of oil necessary for the engine is not easily determined. For ordinary sizes it is from one to four drops per minute, depending on the conditions.

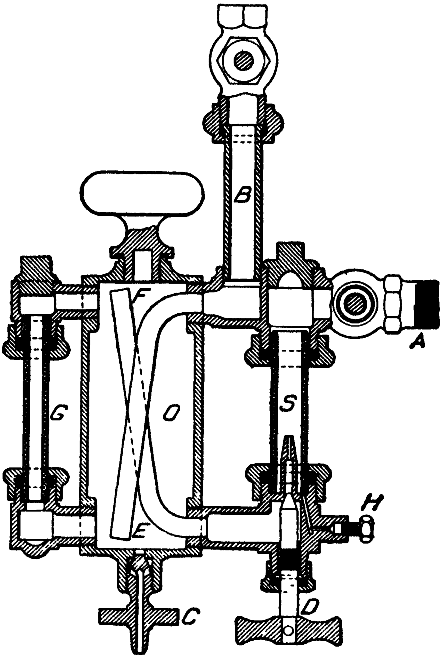


Fig 99. Section of Sight-Feed Lubricator

Instructions for Proper Lubrication. In slow speed engines it is not a difficult matter to attend to the oiling; all the parts are moving slowly and can be readily examined and oiled. Many high speed

engines run so fast that it is impossible to examine the various parts, and special means must be provided for lubrication. It is especially important in high speed engines that there be no heating.

In order to avoid the danger of neglecting to oil a bearing of a high speed engine, it is customary to have all the bearings oiled from one central source. All the oil is supplied to one reservoir, from which pipes lead to all bearings. If this is not done, large oil cups are used so that oiling need not be done so frequently.

In some high speed engines the moving parts are enclosed and the crank runs in a bath of oil. This secures certain oiling and is very effective. All the bearings may be inside this crank case, so that all are oiled in this way. It is thus impossible for a careless operator to overlook one point and so endanger the whole engine.

Large steam engines and turbines are now usually furnished by the manufacturers equipped with some form of oiling system. The large turbines are generally equipped with a continuous oiling system, but in these systems no provision is made for the scientific removal of water and foreign matter from the oil.

Complete Lubrication Systems. We may classify steam turbine oiling and filtering systems as follows:

(1) *Continuous Circulating Systems.* Those in which all the oil used on the bearings is continuously passed through a filter, Fig. 100. Such systems are successful only on small machines.

(2) *Partial Filtration Systems.* Those in which part of the dirtiest oil is continually removed, passed through a filter and returned.

(3) *Batch Filtration Systems.* Those in which all the oil contained in the system is removed and purified in a separate filter, the system being supplied with a fresh batch of clean oil to permit it to operate while the dirty oil is being purified.

Operation of Typical System. The method of operation of the lubricating system of the modern steam turbine is illustrated in Fig. 101. Oil from the different bearings flows by gravity into oil reservoir *B*. A small rotary pump *A*, usually driven from the governor shaft, pumps the oil through the cooler *C*, thence through pipes *D* to the various bearings. The relief valve *L* by-passes any excess oil back into the storage reservoir *B*. In some systems the oil is pumped into an overhead reservoir from which it is allowed to flow by gravity to the bearings. In the illustration shown, the

turbine has four main bearings, *E*, *F*, *G*, and *H*. These bearings are made hollow and are water cooled. The oil is fed into the

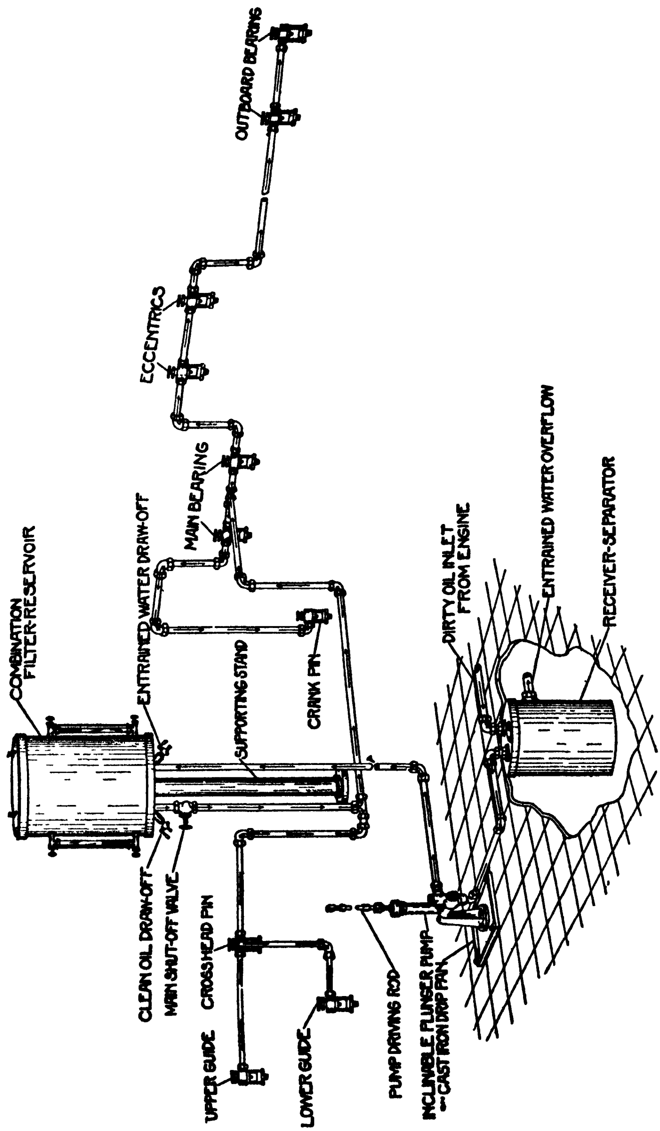


Fig 100 Diagrammatic Drawing of Continuous Oiling and Filtering System for Simple Engine Having Ten Points of Lubrication

bearings at the top and flows out at each end, where it drops down into chambers in the turbine casing and is collected by the

pipe *J* and returned to the reservoir *B*. A screen *K* is provided in the reservoir which removes any large particles of foreign matter.

In such a system there are several places where water finds its way into the oil, the principal place being at the packing gland *M* at the high-pressure end of the turbine casing. Different manufacturers use different forms of packings at this point, but in practically every case some steam escapes which, on coming in contact with the water-cooled bearing at *E*, condenses and mixes with the oil. Again, when turbines are operated with a back pressure, there will be a leakage of condensed steam into the oil at the packing gland *N*, at the exhaust end of the casing. Also the oil cooler *C* and the water-cooled bearings may develop small leaks.

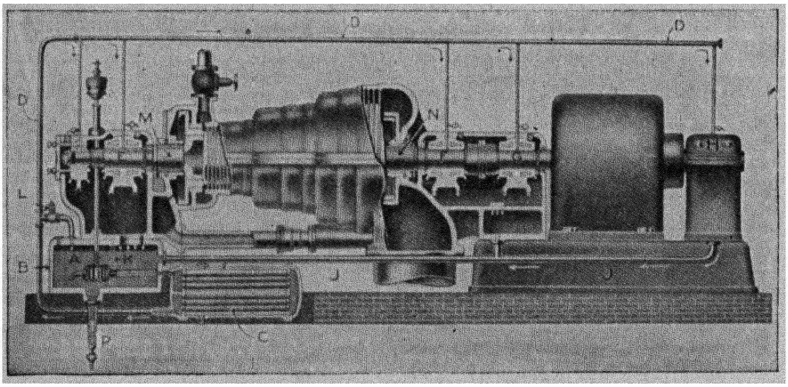


Fig. 101. Section of Modern Steam Turbine, Showing Scheme of Operation of Self-Contained Oil Circulating System.

As a means of getting rid of the water which gets into the system, the cock *P* is provided at the bottom of the reservoir *B*, from which water collecting at this point can be drawn off. The oil usually passes through this tank so rapidly that there is not sufficient time for complete separation of the entrained water.

What has been said concerning the lubrication of large steam turbines applies equally as well to the lubrication of large steam engines.

In Figs. 102 and 103 is shown the plan of operation of a power plant oil filter with the various parts named.

Starting the Engine. Before starting an engine, the oil cups should be started feeding, grease cups screwed down, and the gov-

ernor and other parts of the valve gear oiled. The cylinder lubricator should be started before the engine so that the oil passages will contain oil. The cylinder drain cocks should be open so that any condensed steam in the cylinder will be removed without injury to the cylinder. These precautions having been observed,

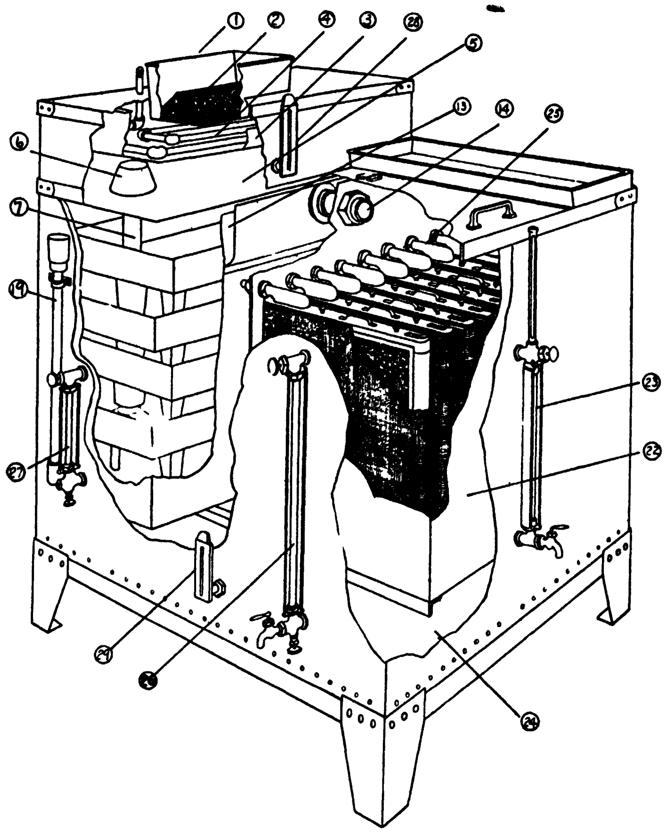


Fig 102 Drawing Showing Interior Construction of Power Plant Oil Filter (See List of Parts under Fig 103)

the throttle may be opened slowly and the engine started and gradually brought to the required speed.

After starting the engine, notice should be taken of the governor and all the lubricating apparatus to see that each is properly performing its function.

When the engine is to operate condensing, the condenser should be started first, if it is in such a position that the water

in the exhaust can drain into it. If the condenser is above the engine and no means are provided for removing the water, the engine should be started non-condensing. When a jet condenser is used, the quantity of injection water should be increased as the load is increased; the amount being determined by the conditions of the vacuum and temperature of the discharge water, which

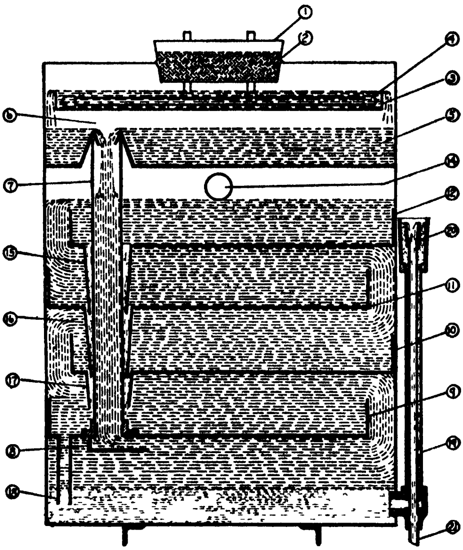


Fig 103 Drawing Showing Section Through Precipitation Compartment and Automatic Water Ejector of Power Plant Oil Filter (See List of Parts)

NAMES OF PARTS OF FIGS 102 AND 103

1—Strainer box for receiving dirty oil, *2*—Removable strainer, *3*—Heating tray, *4*—Steam heating coil, *5*—Dirty oil receiving compartment, *6*—Funnel, *7*—Conductor pipe, *8*—Baffle plate, *9*, *10*, *11*, *12*—Trays, 13—Oil skimmer maintaining constant oil level in precipitation compartment, *14*—Pipe for conducting oil from precipitation compartment to filtering compartment, *15*, *16*, *17*, *18*—Funnels for conducting water direct to bottom of precipitation chamber, *19*—Automatic visible adjustable water overflow pipe, *20*—Overflow funnel, *21*—Water discharge pipe, *22*—Filtering compartment containing non-collapsible cloth-covered filtering units, *23*—Oil gauge for filtering chamber, *24*—Clean oil compartment, *25*—Filtering unit nozzle conducting filtered oil into the clean oil compartment, *26*—Oil gauge for clean oil compartment, *27*—Water gauge for precipitation compartment, *28*—Thermometer showing temperature of oil entering the precipitation compartment, *29*—Thermometer showing temperature of clean oil compartment

should be from 100° to 110° F. If the water is colder than this, it would denote that more injected water is being used than is required.

The foregoing suggestions and indicated precautions are only a few of the more important things that will arise in the course of the erection, setting, and operation of an engine. The one performing

these various duties must at all times exercise good judgment and act according to what his past experiences and that of others have taught under similar circumstances.

ENGINE SPECIFICATIONS

Selecting an Engine. The engineer who has the responsibility of selecting an engine for a given class of service has no small task to perform, if he carefully analyzes all the factors entering into the problem. If the installation contemplated is to be an extensive or expensive one, expert advice should be solicited. Since this is not always to be had, a few suggestions will be given as to how best to proceed when one has to specify an engine for a given service. Consider for the time being that an expert consulting engineer is not available and a rather inexperienced person, or non-technical man, who knows little about the theoretical questions that should be given consideration, has to select the engine. In this case the most satisfactory procedure to follow would be to go to some reliable engine builder and ask him to build or specify an engine that would perform the service required. Having only one builder intrusted, the item of expense would not be chief in his consideration since there would be no competition, therefore the builder would build or specify the best engine possible for the service. If the funds available are limited or must be closely conserved, the intended purchaser may state the limits of cost and then require the builder to come within those limits. It would also be wise on the part of the purchaser to require a guarantee as to the performance of the engine and its maintenance cost for a given period of one year or more.

Drawing Up Specifications. If the purchaser is a competent engineer or he has in his employ such a person, a complete set of specifications may be drawn up and submitted to several engine builders for competitive bids. The specifications submitted should cover in detail the service for which the engine is to be used, the speed at which it is to operate, the type of valves and valve gear desired, the per cent of variation permissible in its governing, and many other items as to the design and detail of construction. Most specifications also specify within what limits the engine must operate, as to the amount of steam used per indicated horsepower per hour, and the range of mechanical efficiency that must be attained. A pro-

vision should be made in the contract as to the conditions under which the acceptance test will be made and by whom.

The form of specification usually submitted by the builders and which in general will be like those written by an engineer when requesting bids, is submitted herewith. This may be taken as a typical specification, the items being changed to meet different conditions of service as the particular case demands.

SPECIFICATIONS OF A VERTICAL CROSS-COMPOUND, SIDE-CRANK, ENGINE, ARRANGED FOR 1000-K W DIRECT CONNECTED GENERATOR, 60 CYCLE ALTERNATOR

Size, Power, and Dimensions

Diameter of high pressure cylinder, 27 inches.
Diameter of low pressure cylinder, 54 inches.
Stroke, 42 inches.
Revolutions per minute, 120.
Initial steam pressure, 125 pounds, 26 inches vacuum, condensing.
Rated load in indicated horsepower, 1,520; cut-off, 26/100
At ½ cut-off, indicated horsepower, 2,100, maximum cut-off, 7/10
Estimated total weight of engine, 346,000 pounds
Weight of wheel, 92,000 pounds. Diameter, 16 feet Face, —— inches.
Diameter of bearings, 19 inches Length, 35 inches.
Diameter of shaft between bearings, 22 inches
Diameter of crank pin, 9 inches. Length, 8 inches
Diameter of crosshead pin, 8 inches Length, 8 inches.
Bearing surface of crosshead, 17 inches by 20 inches.
Diameter of piston rod, 5 inches
Diameter of throttle valve, 12 inches.
Diameter of exhaust opening, 22 inches.

Workmanship and Materials

The workmanship, finish, fitting, and materials will be first-class in every particular All forgings will be of open-hearth steel or hammered iron, as hereafter specified All castings subject to wear, such as cylinders, guides, pistons, etc , will be poured from mixtures containing charcoal iron, graded according to the size of casting in order to secure the proper hardness and closeness of grain.

The engine will be made to gauge and interchangeable This feature will be thoroughly carried out.

Flat surfaces will be scraped to surface plates, and surface and cylindrical grinding will be used where advantageous.

Guarantee

We guarantee the workmanship and materials in the engine to be first-class and in fulfillment of our guarantee we will give a duplicate to take the place of any part that may prove defective in material, workmanship, or design within one year after the engine is started.

We guarantee the engine to regulate from no load to full rated load within 2 per cent variation of speed

We guarantee the engine to run in a smooth and proper manner without undue heating or vibration.

CYLINDERS

The cylinders and steam chests will be neatly covered with sheet iron lagging, enclosing a thick layer of the best quality of asbestos or magnesia fiber The cylinder and steam chest covers will also be provided at each end with thin iron castings or covers. The cylinders will be provided at each end with a patent combination relief-valve and drip-cock of large diameter, adjustable to open automatically at any pressure desired Being operated by hand as drip-cocks, these will not stick or become inoperative from disuse, but will relieve dangerous pressure from water or other causes.

JACKETS AND RECEIVER

The high pressure cylinder will be steam-jacketed and there will be a receiver of large capacity between cylinders.

The receiver will be filled with seamless brass heating coils containing steam at boiler pressure. The high pressure jacket and these coils should be piped in series, so that steam will pass through in the order named, and since the steam in the low pressure coils is hotter than the receiver steam, the latter will be considerably superheated upon entering the low pressure cylinder, and enough of the former will be condensed in the coils to cause brisk circulation in the high pressure cylinder jacket which is necessary to its efficiency It is the aim of this arrangement to keep the steam dry throughout its course through the engine without the loss of any portion of heat of the jacket to the exhaust steam. The water condensed in jackets and in the coils should be returned to the boiler.

VALVES

Both cylinders will be four-ported and provided with valves of the flat gridiron type of our standard form

The valves slide crosswise of the cylinder upon gridiron seats, which are separate and removable from the cylinder itself. Since the valves are of the gridiron type, a very small stroke is necessary to give full opening, and they move with an intermittent motion, standing still when closed, and only require power to operate when open and relieved of steam pressure. The clearance is reduced to about one-half of that necessary with valves of the Corliss type

These valves possess the following advantages:

They give rapid opening of port with the least amount of wear and power required to operate

The clearance space is reduced to a minimum.

They will not stick when the engine is started, and are easy to keep lubricated

They wipe over and wear evenly, are unbalanced, and hence *will be tight* when old as well as when new.

VALVE GEAR

The main valves will be driven by a fixed eccentric controlling the admission of the steam and the opening and the closing of the exhaust. The cutting off of the steam will be effected by the cut-off valves which are controlled by the governor.

The valve gear is positive, composed of simple levers and links, and the cut-off can take place at any point between zero and the maximum cut-off. The cut-off, except at light loads, occurs when the main and cut-off valves are moving in opposite directions, and the cut-off is as sharp as with a releasing type of valve gear notwithstanding the short stroke used

The cut-off is varied simultaneously upon all the cylinders in such a manner that the work done in each is approximately equal, as is also the drop in temperature of steam in each This adds to smooth running and gives best distribution of steam for economy at all cut-offs under variable loads

The valve gear will be constructed in the most substantial and durable manner, and in such a way as to equalize the cut-off at both ends of the cylinders for all cut-offs Rock-shafts, pins, and links will be made of open-hearth steel. Connecting links will be fitted with bronze ends having quick taper key adjustment The eccentric straps will be lined with babbitt hammered in and bored out The rock-shaft bearing will be babbitted and adjustable.

Governors

The governor will be situated on the main shaft of the engine A change in position of the centrifugal weights revolves the eccentric controlling the position and motion of the cut-off valves around the shaft and varies the point of cut-off

All the bearing pins in the governor will be made of tool steel hardened and ground, turning in bearings bushed with phosphor bronze The centrifugal force of each governor weight is resisted by a plate spring through a pin having hardened steel points resting in phosphor bronze cups, one at the end of the spring and the other at the center of gravity of the governor weight. The centrifugal force of the governor weights is thus opposed in a direct and frictionless manner without causing pressure or friction on the pins upon which the governor weights swing This governor will regulate the speed of the engine with a closeness and certainty impossible with a fly-ball governor, and its action is unaffected by wide and sudden fluctuations of load. The governor will control both cut-off eccentrics.

Pistons, Piston Rods, and Stuffing Boxes

The pistons will be cored out and provided with internal ribbing, making them very light and strong They will be secured to the piston rod by being forced upon a taper, with shoulder beyond, and by a nut, with a simple but efficient locking device. The pistons will be provided with cast-iron packing rings

The piston rods will be of open-hearth steel running through deep stuffing boxes and babbitted glands The rods will not touch the heads—which will be bored large—bronze rings fitting the rods in the bottom of each stuffing box and preventing escape of packing to the interior of the cylinder.

Low pressure piston will be of steel.

Framing

This will consist, for each cylinder, of a deep and massive base containing the main bearings. On the back of each base will stand a very heavy rectangular column, as shown in the blue print, securely bolted to a heavy frame head In front the frame heads will be connected to the bases by forged steel columns bolted by flanges forged solid with the columns. The

rear column will support the cylinders when the forged columns in front are removed, facilitating the placing of shaft and other parts.

GUIDES, CROSSHEADS, AND CROSSHEAD PINS

The guides will be separate from the frame and adjustable for wear with an oil dish at the bottom which, together with a thin brass fringe upon the bottom of the crosshead, forms an efficient self-oiling device.

The crossheads will be of open-hearth steel fitted with babbitted cast-iron shoes.

The crosshead pins will be of open-hearth steel flattened on two sides to prevent wearing oval

CONNECTING RODS AND BOXES

The connecting rod will be of forged steel, provided with gib and key ends The straps will be provided with pinching bolts which will prevent spreading Both crank and crosshead pin boxes will be lined with babbitt hammered in and bored out

The body of the connecting rod will be made of larger section than the piston rod, being designed properly for the added strain due to its length and angular motion

SHAFT, CRANK PIN, AND DISK

The shaft will be piled and faggoted hammered iron forging.

The crank disk will be made with counterbalance, of a mixture containing charcoal iron. The crank pin will be made of forged steel. The shaft and crank pin will be forced into the disk by hydraulic pressure and the disk will also be keyed securely to the shaft

MAIN BEARINGS AND REMOVABLE SHELLS

The main bearings will be fitted with cylindrical shells, lined with babbitt, hammered in and bored out. These shells can easily be taken out by removing the cap and simply jacking up the shaft sufficiently to take the weight off the bearings, when they can be revolved around the shaft and taken out without disturbing any other parts of the engine The shells are made hollow for water circulation. This is not intended to be used ordinarily, but in case dirt or other unusual conditions should cause the bearing to heat, it often enables the engine to complete its run without stopping.

The main bearings will be provided with a self-oiling device which will keep them flooded with oil

OIL FEED SYSTEM

The feed will be positive and adjustable and the system will be closed, so that there will be little waste and deterioration of oil Rings at the ends of the bearings will throw off escaping oil into close-fitting shields with suitable drain pipes leading to a large settling reservoir beneath A small pump driven from the valve gear will deliver the oil to a feed tank at each bearing This tank will be provided with an adjustable feed outlet pipe leading to the bearings, and with a gauge-glass and by-pass overflow, and can be filled by hand and used as an ordinary oil cup if it is desired to cut off the automatic supply while the engine is running.

FLYWHEEL

The wheel will be cast in halves and will be bolted together at the hub with reamed bolts carefully fitted in holes drilled from the solid, and the parts will be planed where they join. Steel arrow head links will be used

at the rim. The wheel will be carefully designed throughout in order to have a large factor of safety, and both edges and face of rim will be turned true

PLATFORMS

Platforms convenient for handling and operating the engine will be provided as shown in print. These can be arranged to suit the location of the engine and will be made stiff to avoid vibration. The hand railings will be of seamless brass tubing, fitted into brass caps or iron posts. The platform plates will be diamond figured, planed where they join together and neatly fitted Stairs will be made of channel iron, with cast-iron diamond threads

FIXTURES

The following fixtures will be provided: throttle valve; indicator motion; complete outfit of sight-feed cylinder lubricators; glass body oil pumps; grease cups for valve gear; centrifugal crank pin oilers; reservoirs with sight-feed outlets; oil pipes and wipers for oiling the main parts of the engine conveniently and continuously; relief valves for each end of the cylinders; drip-cocks; wrenches, foundation bolts; and foundation plans.

Contract. After the engine has been selected and the builders determined, a written contract should be entered into in order to make it a legal document. A contract, according to Blackstone, is an agreement upon sufficient consideration to do or not to do a particular thing. In the case of the purchasing of an engine, the builder agrees to build, erect, and put into operation an engine in accordance with the specifications and drawings submitted, which items become a legal portion of the contract. The purchaser may also require that the engine be ready for operation in a given time and that it must also come up to certain requirements in its performance, as previously mentioned. In consideration of the foregoing, the purchaser agrees to pay the builders a specified sum of money, either in one payment or more as determined by them. The wording and statement of the contract should be carefully prepared, in order to avoid any possible misinterpretation of any of its provisions.

COST OF ENGINES AND OF THEIR OPERATION

The question of the cost of an engine and of its erection and operation is indeed a very vital one. This cost can not be classified in a brief way, since there are so many contributing factors that differ widely in different localities. For example, no well-defined indication of the cost of operation can be given, and the cost of labor and material are fluctuating items of expense; therefore, the cost of the engine can not be stated definitely, since in a brief interval of time it may be considerably more or less. Many articles appear

TABLE III

Price of Single Cylinder Corliss Engines, Set and Erected

Size of Cylinder, Inches	Horse-power	Cost of Engine	Cost of Foundation	Cost of Erecting	Cost of Piping	Total Cost
16×36	125	$1950	$325	$210	$180	$2665
18×36	155	2150	375	240	200	2965
18×48	200	2600	425	260	220	3505
20×48	230	2850	525	275	250	3900
22×42	250	3000	550	300	310	4160
24×48	320	4000	700	375	390	5465
28×48	425	5150	900	500	800	7650
30×48	490	5800	1200	600	1070	8670

from time to time in the leading engineering papers which give valuable information upon such matters and usually this information is correct since it is given currently with the ascertained cost of various items. It is, therefore, suggested that if the latest and perhaps most authentic information is desired upon these items of expense that such articles as appear in the papers mentioned should be consulted.

Engine Costs. As an indication of what such expense will be Tables III and IV, as devised by Dean C. H. Benjamin, are given.

TABLE IV

Cost of High Speed, Single Cylinder Engines

Horsepower	Size of Cylinders Inches	Steam Pressure Pounds Per Square Inch	R P M	Cost Del F O B	Cost of Sub-Base	Cost of Engine Foundation	Cost of Superintendence—Labor	Cost of Handling	Total Cost (Engine set up on Foundation)
50	9×10	100	300	$695	$45	$65	$70	$10	$885
75	10×12	100	300	890	50	75	70	15	1100
100	12×12	100	290	1085	50	80	70	15	1300
125	13×14	100	275	1260	70	95	70	17	1512
150	15×14	100	245	1595	80	110	75	20	1880
	14×16								
200	18×16	100	225	2010	90	140	85	25	2350
250	19×18	100	200	2800	250	200	100	35	3385

Relative Cost of Operation Items. The cost of the operation of a steam plant is properly made up of several items, viz, rent or

TABLE V

Cost of Installation and Operation for One Year

Kind of Engine	Total Cost Engines and Boilers	Annual Cost of Both Engine and Boilers, Depreciation and Interest	Total Tons Coal Per Year of 3000 Hours	Lubricants	Labor, Engine and Boilers	Cost of Power, Coal at $2 Per Ton, 1 Year
Simple Slide Valve Non-condensing	$29 75	$4 03	6750	$1 02	$5 00	$23 55
Compound Slide Valve Non-condensing	31 50	4 38	5660	1 25	4 50	21 45
Compound Slide Valve Condensing	29.80	4 26	4050	1 25	3 80	17 41
Simple Corliss Non-condensing	32 25	3 84	6075	1 00	4 70	21 00
Compound Corliss Condensing	30 87	3 76	3375	1 25	3 50	16 25
Triple Corliss Condensing	34 25	4.28	3110	1 50	4 00	16 00

interest on real estate; interest on investment; maintenance, etc., of equipment; fuel; water; supplies; and attendance.

The relative value of these various items for a large central station lighting plant was given in the *Engineering Magazine*, May, 1905. Taking the total cost of maintaining the station as 100 per cent, the following were the average costs of the various items: Fuel 52.5%; wages 26.4%; water 2.2%; oil and waste 1.8%; rent 4.35%; station repairs 2.2%; steam repairs 5.45%; electric repairs 5.1%.

Annual Operation Expenses. Professor Carpenter in the *Economist* summarizes the cost of installation and the operation of an entire plant for one year of 3,000 hours as given in Table V. A coal consumption of 4.5 pounds per boiler horsepower per hour is assumed and the cost given per engine horsepower is for a 1,000 horsepower engine.

An illustration involving the items given in Table V will serve to make it clearer. The case of a simple slide valve non-condensing plant will be considered.

Cost of engines and boilers at $29.75 per horsepower = $29,750
Annual cost of depreciation and interest at $4 03 per h p. = $4,030
Annual cost of coal at $2.00 per ton = 6750×2 = $13,500.
Annual cost of lubricants at $1.02 per horsepower = $1 02×100 = $1,020.
Annual cost of labor at $5 00 per horsepower = $5.00×1000 = $5,000.

Annual cost for the last four items = $23,550 or $23.55 per h p.

The foregoing tables will serve to give some idea of the cost of engines, also of the cost of operation of a steam plant, but it must be remembered that the figures given will not be exact for all localities or for all times, due to the changing influences previously mentioned.

ENGINE TESTS

Importance of Tests. It was mentioned in connection with the discussion of specifications and contracts that often a guarantee is given by the builder as to the economical performance of a steam engine, hence it is required that the engine be tested in order to ascertain whether or not it meets the provisions of the guarantee. While this is one reason that may be assigned for testing an engine, yet there are several others of importance. The user from time to time may want to ascertain the condition of the engine as a whole and also the condition of particular features such as the valves, etc. For purely theoretical reasons an engine is often tested in order that an analytical study may be made of its performance under various conditions and in comparison with other engines of different classes. Many such tests have resulted in obtaining data, the facts of which have demonstrated to both the builder and the user possible economies. Because of the information thus obtained, the builder has been enabled to design a better engine, and the user to operate his engine more advantageously. The remarks given will suffice to indicate that the ultimate object of an engine test is the determination of the *economy* with which the engine produces a given amount of *power*. In steam engines the economy, as usually ascertained, relates to the weight of steam consumed, to the quantity of coal required to make the steam, or to the number of heat units supplied. The elementary quantities concerned are accordingly two in number, viz, the amount of steam, fuel, or heat (as the case may be) consumed, and the amount of power developed. How to determine these quantities is the problem.

A. S. M. E. Code. The American Society of Mechanical Engineers (A.S.M.E.) deemed the testing of engines according to some definite and standard method of such importance that a committee was appointed to devise a standard code. This, after much labor and diligent study, was presented to the Society and adopted. The full report appears in Volume 24 (1904), page 713, of the Trans-

actions. Since the report of the first committee, various committees have studied the Code and offered revisions which were ultimately adopted in 1915. In so far as the conditions will permit, this code should be followed. The 1915 Code for Conducting Tests of Reciprocating Steam Engines is too lengthy to give in its entirety but is summarized as follows:

Method of Conducting Steam Engine Tests
Code of 1915

INSTRUCTIONS REGARDING TESTS IN GENERAL

(1) *Object.* Ascertain the specific object of the test, and keep this in view not only in the work of preparation but also during the progress of the test, and do not let it be obscured by devoting too close attention to matters of minor importance. Whatever the object of the test may be, accuracy and reliability must underlie the work from beginning to end.

If questions of fulfillment of contract are involved, there should be a clear understanding between all the parties, preferably in writing, as to the operating conditions which should obtain during the trial, the methods of testing to be followed, corrections to be made in case the conditions actually existing during the test differ from those specified, and as to all other matters about which dispute may arise, unless these are already expressed in the contract itself.

Among the many objects of performance tests, the following may be noted: Determination of capacity and efficiency and how these compare with standard guaranteed results, comparison of different conditions or methods of operation; determination of the cause of either inferior or superior results; comparison of different kinds of fuel; and determination of the effect of changes of design or proportion upon capacity or efficiency, etc.

(2) *Preparations.* Measure the dimensions of the principal parts of the apparatus to be tested, so far as they bear on the objects in view, or determine these from correct working drawings. Notice the general features of the apparatus, both exterior and interior, and make sketches, if needed, to show unusual points of design.

The dimensions of engine cylinders should be taken when they are cold, and, if extreme accuracy is required, as in scientific investigations, corrections should be applied to conform to the mean working temperature. If the cylinders are much worn, the average diameter should be found. The clearance of the cylinders may be determined approximately from working drawings of the

engine. For accurate work, when practicable, the clearance should be determined by the water measurement method.

Make a thorough examination of the physical condition of all parts of the plant or apparatus which concern the object in view, and record the conditions found, together with any points in the matter of operation which bear thereon.

Ascertain the interior condition of all steam, air, gas, or water cylinders and the condition of their pistons, and of water plungers and impellers, together with the valves and valve seats belonging thereto. Examine for air leaks in exhaust piping, condenser, packings, etc., by using the vacuum-gage or candle-flame test or by filling the piping, etc., with warm water under a slight head. Examine steam, air, gas, or water piping, traps, drip valves, blow-off cocks, safety valves, relief valves, heaters, etc., and make sure that they do not leak.

If the object of the test is to determine the highest efficiency or capacity obtainable, any physical defects or defects of operation tending to make the result unfavorable should first be remedied, all fouled parts being cleaned and the whole put in first-class condition. If, on the other hand, the object is to ascertain the performance under existing conditions, no such preparation is either required or desired.

In steam tests make sure that there is no leakage through blow-offs, drips, etc., or through any steam or water connections of the plant or apparatus undergoing test which would in any way affect the results. All such connections should be blanked off, or satisfactory assurance should be obtained that no leakage is going on either out or in. This is a most important matter, and no assurance should be considered satisfactory unless it is susceptible of absolute demonstration.

(3) *Apparatus and Instruments.* Select the apparatus and instruments specified later, locate and install the same, and complete the preparations for the work in view. The arrangement and location of the testing appliances in every case must be left to the judgment and ingenuity of the engineer in charge, the details being largely dependent upon locality and surroundings. One guiding rule, however, should always be kept in view, viz, *see that the apparatus and instruments are substantially reliable and arranged in such a way as to obtain correct data.* The following is a list of apparatus and instruments needed, together with a description of their leading features, methods of application and use, and, where needed, methods of calibration.

Weighing Scales. For determining the weight of coal, oil, water, etc, ordinary platform scales serve every purpose Too much dependence, however,

should not be placed upon their reliability without first calibrating them by the use of standard weights and carefully examining the knife-edges, bearing plates, and ring suspension to see that they are all in good working order.

Water Weighing and Measuring Apparatus In tests of complete steam power plants, where it is required to measure the feed water without unnecessary change in the working conditions, a water meter may be employed Meter measurement may also be required in many other cases such as locomotive and marine service. The accuracy of meters should be determined by calibration in place under the conditions of use.

If a large quantity of water is to be measured, an automatic water weigher, a rotary, disk, or Venturi meter, a weir, or some form of orifice measurement may be employed. In any case the measuring apparatus should be calibrated under the conditions of use, unless its design is such that standard formulas and constants may be applied for determining the discharge If recording mechanism is employed in connection with orifice or weir measuring apparatus, make sure its record is reliable.

In measuring jacket water or any supply under pressure which has a temperature exceeding 212° F , the water should first be cooled, which may be done by discharging it into a tank of cold water previously weighed or by passing it through a coil of pipe submerged in running and colder water, thereby preventing loss by evaporation, which occurs when such hot water is discharged into the open air If such water is untrapped, the drain pipe should be provided with a gage glass and the outlet choked, so as to keep the water in sight in the glass

Venturi meters, Pitot tubes, pitometers, and orifices may be used for measuring water discharged by pumps through pipes under pressure

Steam-Measuring Apparatus Various forms of steam meters may be employed for measuring steam, provided that such meters are properly calibrated under conditions of use and that the pulsations of pressure, if any, are not serious For measuring the steam used by the auxiliaries of a steam plant, either individually or collectively, the orifice form of steam meter may be used, consisting of an orifice in a plate inserted between the two halves of a pair of flanges in the pipe through which the steam passes or placed in a by-pass through which the steam is diverted, with gage pipe on either side for determining the fall in pressure The quantity of steam represented by the various differences of pressure which occur may be found by arranging the apparatus so as to draw steam through the orifice and discharge it into a tank of water resting on platform scales, by which its actual weight in a given time is determined

A plate $\frac{1}{8}$ inch thick containing an orifice 1 inch in diameter, with square edges, will discharge the approximate quantities of dry steam per hour given in Table VI with various pressure drops, the pressure below the orifice being 100 pounds by gage

The water-glass method affords an approximate means for determining the steam consumption of auxiliaries and for measuring the leakages of steam and water from the boiler and its connections This method consists of shutting off all secondary feed valves (which must be known to be tight) and the main feed valve, thereby stopping absolutely the entrance or exit of water at the feed pipes to the boiler; then maintaining the steam pressure (by means of a very slow fire) at a fixed point, which is approximately that of the working pressure, and observing the rate at which the water falls in the gage glasses. It is well in this test, as

TABLE VI

Discharge through Orifice 1 Inch in Diameter at 100 Pounds Pressure

Pressure Drop (lb per sq in)	Dry Steam (lb per hr)
½	430
1	615
2	930
3	1200
4	1400
5	1560
10	2180
15	2640
20	3050

in other work of this character, to make observations every ten minutes and to continue them until the differences between successive readings attain a constant rate In many cases the conditions will have become constant at the expiration of fifteen minutes from the time of shutting the valves, and thereafter the fall of the water due to leakage of steam and water becomes approximately constant It is usually sufficient, after this time, to continue the test for two hours, thereby obtaining a number of half-hour periods When this test is finished, the quantity of leakage is ascertained by calculating the volume of water which has disappeared, using the area of the water level and the depth shown on the glass, making due allowance for the weight of one cubic foot of water at the observed pressure The water columns should not be blown down during the time a water-glass test is going on nor for a period of at least one hour before it begins

If there is opportunity for condensation to occur and collect in the steam pipe during the leakage test, the quantity should be determined as closely as desirable and properly allowed for

Pressure Gages For determining pressure, the gages belonging to the plant may be used, provided they are compared with a standardized gage of the spring or mercury type and verified, due allowance being made for the head of water, if any, standing in the connecting pipe Such comparisons should be made with both gages at their respective normal temperatures In the use of spring gages for steam, the gages should be protected by proper siphons of water seals and no leakage should be allowed at the gage cock The gages should also be located so that they will not be unduly heated.

For measuring low pressure or vacuums, a U-tube gage may be employed or a spring gage may be used, provided it is referred to a standard and corrected for water in the connecting pipe In cases where extreme vacuums are to be measured, as in turbine practice, the absolute-pressure gage is useful, provided the exhaustion is complete and no air is admitted afterward

For determining steam pressure on the two sides of an orifice, two gages should be used which are carefully graduated to single pounds or, better, one gage should be used and should be piped up so as to connect at will to either side of the orifice. A differential gage may also be employed, indicating at once

the pressure drop If the pressure drop is small, a glass U-tube containing mercury may be used If the drop is less than one-half pound, water columns may be substituted.

For determining the water pressure in the force main of a pumping engine, the gage should be one which is sensitive to changes amounting to one-half per cent of the pressure indicated If such a gage is not a part of the equipment of the plant, a special test gage should be attached

For calibrating gages indicating pressures above the atmosphere, the dead-weight testing apparatus, which is manufactured by many of the prominent gage makers, may be employed as a standard of comparison It consists of a vertical plunger nicely fitted to a cylinder containing oil or glycerin, through the medium of which the pressure is transmitted to the gage. The plunger is surmounted by a circular stand on which weights may be placed and by means of which any desired pressure can be secured. The total weight, in pounds, on the plunger (including weight of plunger) divided by the average area of the plunger and of the bushing which receives it, in square inches, gives the pressure in pounds per square inch

Another standard of comparison is the mercury column If this instrument is used, assurance must be had that it is properly graduated with reference to the ever-varying zero point, that the mercury is pure; and that the proper correction is made for any difference between the actual temperature and the temperature at which the instrument was graduated

For pressures below the atmosphere, an air pump or some other means of producing a vacuum is required and reference must be made to a mercury gage Such a gage may be a U-tube having a length of 30 inches and with both arms properly filled with pure mercury

Thermometers Thermometers should be of the kind having graduations marked on the glass stem Those used for temperatures above 500° F should have nitrogen in the top of the bore They should also have a small safety bulb at the top Thermometers constructed in this way can be used satisfactorily up to 1000° F.

Thermometers which are used for important data should be calibrated before and after a test by reference to standard thermometers They may be calibrated, if desired, by direct comparison with standard thermometers certified by the U S Bureau of Standards

A thermometer well consists of a hollow plug threaded at the upper end and screwed into a threaded hole in the top of a horizontal pipe, the lower part extending vertically into the interior of the pipe as far as the center, if practicable The inside diameter of the thermometer tube and the well should be filled with mercury or high-grade mineral oil for temperatures below 500 degrees and with soft solder for higher temperatures For superheated steam the immersed portion should be fluted so as to increase the area of the absorbing surface

When the stem is not immersed, the correction to be added to its reading is 0 000088 $n(T-t)$, in which n is the number of degrees on the scale not immersed, T the indicated temperature, and t the mean temperature of the air surrounding the stem, as shown by a thermometer suspended at the mean point.

For accurate work thermometers should be standardized for the immersion at which they are intended to be used, and such immersion should be recorded.

For ordinary work thermometers may be used without correction if they are of the type that are graduated at a given immersion, the degree of immersion

being marked on the stem and the temperature of the exposed stem being approximately that at which it was graduated.

Barometers. For important or extremely accurate steam tests and gas engine tests, the pressure of the atmosphere should be taken either by a mercurial or aneroid barometer and the reading from this instrument, reduced to pounds pressure per square inch, should be employed in determining the absolute steam pressure. In many cases it is sufficient to refer to the daily records of the nearest station of the Government Weather Bureau. These records, which refer to sea level, should be corrected for altitude Aneroid barometers may be readily calibrated by comparing them with a mercury barometer, making proper temperature corrections.

Steam Calorimeters The most satisfactory instruments for determining the amount of moisture in steam are calorimeters that operate on the throttling principle or that combine the throttling and separating principles, the orifice used being of such size as to throttle to atmospheric pressure and the instrument being provided with two thermometers, one showing the temperature above the orifice and the other that below it If no commercial make of calorimeter is available on a test, an instrument of the throttling type can be made of pipe fittings Instruments working on the separating principle alone may also be employed, also certain forms of electric calorimeters

In using a steam calorimeter great care must be exercised in attaching the instrument to a properly located sampling tube or pipe The sampling tube, pipe, or nozzle should be made of ½-inch iron pipe and inserted in the steam main at a point where the entrained moisture is likely to be most thoroughly mixed The inner end of the pipe, which should extend nearly across to the opposite side of the main, should be closed and the interior portion perforated with not less than twenty ⅛-inch holes equally distributed from end to end and preferably drilled in irregular or spiral rows, with the first hole not less than ½ inch from the wall of the pipe.

If it is necessary to attach the sampling nozzle at a point near the end of a long horizontal run, a drip pipe should be provided a short distance in front of the nozzle, preferably at a pocket formed by some fitting, and the water running along the bottom of the main drawn off, weighed, and added to the moisture shown by the calorimeter; or, better, a steam separator should be installed at the point noted

Indicators To determine the amount of power developed in the cylinder of a reciprocating engine or that expended in a pump or compressor cylinder, the instrument required is the steam engine indicator One or more of these instruments are attached to the cylinder or cylinders and operated from the crosshead or main shaft by the use of proper driving rig. As to the selection of the make of instrument, it should be one which is in all respects of first-class construction and is adapted to the purpose for which it is to be used

Outside spring indicators are preferred for superheated steam and in other cases where the temperature of the gas or vapor is very high or low

Planimeters To determine the area of indicator diagrams, from which to ascertain the mean effective pressure, it is convenient to use some form of planimeter The simplest, and probably the most desirable, instrument is the Amsler polar planimeter, in which the area is registered in square inches

It is desirable to calibrate a planimeter from time to time by running it over

a figure having a known area, such as a right-angle triangle of, say, 4 inches in length and 2 inches in height, observing whether it checks with the computed area

Tachometers and Other Speed-Measuring Apparatus For determining the speed of revolution of an engine shaft, especially where the speed exceeds 300 r.p m , a convenient instrument is a tachometer which continually indicates on a dial the number of turns per minute. This instrument can be arranged to have a permanent location and to be operated continuously when the engine is running, or it can be a portable instrument which is held in the hand and applied for the time being to the end of the shaft These instruments are of four general classes, viz, fly-ball, liquid, electromagnetic, and vibration

These instruments should be calibrated by comparison with the record obtained by counting with the watch and a speed recorder or indicator the number of turns per minute.

The determination of variation of speed during a single revolution, or the effect due to sudden changes of the load, is desirable, especially in engines driving electric generators used for lighting purposes There is no recognized standard method of making such determinations, and if they are desired the method may be devised to suit the requirements

One method* for determining the instantaneous variation of speed which accompanies a change of load is described as follows A screen containing a narrow slot is placed on the end of a bar and vibrated by means of an electric current A corresponding slot in a stationary screen is placed parallel to, and nearly touching, the vibrating screen, the two screens being placed a short distance from the flywheel of the engine in such a position that the observer can look through the two slots in the direction of the spokes of the wheel The vibrations are adjusted so as to conform to the frequency with which the spokes of the wheel pass the slots When this is done the observer viewing the wheel through the slots sees what appears to be a stationary flywheel When a change in velocity of the flywheel occurs, the wheel appears to revolve either backward or forward according to the direction of the change By careful observations of the amount of this motion, the angular change of velocity during any given time is ascertained

Friction Brakes or Absorption Dynamometers The power delivered by an engine may be determined by the application of a Prony brake to the rim of the flywheel The friction device may consist of a simple band or rope, a number of ropes, or a series of blocks encircling the wheel Weighing scales either of the platform or spring type are required for measuring the torque For long runs the wheel is made with interior flanges for holding water to keep the rim cool

The most satisfactory brake for absorbing and measuring power is some form of water friction brake The advantage of a water brake is that it can be employed equally well for large or small amounts of power, and it is necessarily kept cool by the water upon which it depends for its operation With this brake the determination of the quantity of water used and the number of degrees its temperature is raised (when corrected for radiation) furnishes a means of computing the amount of heat converted into work and thereby obtaining an additional measurement of power developed

Another satisfactory form of brake is the electric dynamometer, in which the work is transformed into electric energy and the torque is measured in the same manner as in a Prony brake

Transmission Dynamometers Transmission dynamometers furnish means for determining the amount of power delivered by an engine under working conditions. In the case of a mill engine it is the power transmitted from the main shaft of the engine to the shaft of the mill. If this power is carried through a belt, the dynamometer measures the net amount of force transmitted In a marine engine driving a screw propeller through a long shaft, the dynamometer shows the torsional strain on the shaft at a point as near as practicable to the engine In a locomotive the dynamometer measures the amount of pull on the drawbar through which the power is transmitted to the first car of the train

Steam Tables Quantities depending upon the properties of saturated and superheated steam, which are used throughout the Code, such as B t u per pound of steam, and temperatures corresponding to various pressures, etc , are based on Marks and Davis' tables (edition of 1909) The report of a test should state the authorship of the tables on which the calculations are based

(4) *Miscellaneous Instructions.* The person in charge of a test should have the aid of a sufficient number of assistants, so that he may be free to give special attention to any part of the work whenever and wherever it may be required. He should make sure that the instruments and testing apparatus continually give reliable indications and that the readings are correctly recorded. He should also keep in view, at all points, the operation of the plant or part of the plant under test and see that the operating conditions determined on are maintained and that nothing occurs, either by accident or design, to vitiate the data. This last precaution is especially needed in guarantee tests.

Before a test is undertaken, it is important that the engine shall have been in operation a sufficient length of time to attain working temperatures and proper operating conditions throughout, so that the results of the test may express the true working performance. An exception should be noted where the object of the test is to obtain the working performance, in which case all the conditions should conform to those of regular service.

In preparation for a test to demonstrate maximum efficiency, it is desirable to run preliminary tests for the purpose of determining the most advantageous conditions.

(5) *Operating Conditions.* In all tests in which the object is to determine the performance under conditions of maximum efficiency or in which it is desired to ascertain the effect of predetermined conditions of operation, all such conditions which have an appreciable effect upon the efficiency should be maintained as nearly uniform during the trial as the limitations of practical work will permit.

On the other hand, if the object of the test is to determine the performance under working conditions, no attempt at uniformity

is either desired or required unless this uniformity corresponds to the regular practice; the usual working conditions should, therefore, prevail throughout the trial.

(6) *Records.* A log of the data should be entered in notebooks or on blank sheets suitably prepared in advance. This should be done in such manner that the test may be divided into hour periods or, if necessary, periods of less duration and the leading data obtained for any one or more periods as desired, thereby showing the degree of uniformity obtained.

The readings of the various instruments and apparatus concerned in the test other than those showing quantities of consumption (such as fuel, water, etc.), should be taken at intervals not exceeding one-half hour and entered in the log. Whenever the indications fluctuate, the intervals should be reduced according to the extent of the fluctations. When it is essential that a number of instruments be read simultaneously, there should be an observer stationed at each one and the readings should be taken on a signal from a timekeeper

Make a memorandum of every event connected with the progress of the test, however unnecessary it may appear at the time. A record should be made of the exact time of every such occurrence and the time of taking every weight and observation. For the purpose of identification the signature of the observer and the date should be applied to each log sheet or record.

In the simple matter of weighing water by the tank-full, which is required in many tests, a series of marks, or tallies, should never be trusted. The time each tank is emptied should be recorded. Such work should not be delegated to unreliable assistants and, whenever practicable, one or more men should be assigned solely to that work.

(7) *Plotting Data and Results.* If it is desired to show the uniformity of the data at a glance, the whole log of the trial should be plotted on a chart, preferably while the test is in progress, using horizontal distances to represent times of observation and vertical distances on suitable scales to represent various data as recorded.

RULES FOR CONDUCTING TESTS OF RECIPROCATING STEAM ENGINES

(1) *Introduction.* This code for steam engine tests applies to tests for determining the performance of the engine alone (including reheaters and jackets, if any), apart from that of steam-driven auxiliaries which are necessary to its operation. For tests of engines and auxiliaries combined and for tests of multiple-expansion engines from which steam is withdrawn for heating feed water

or otherwise, a separate and distinct code was prepared by the Power Test Committee of the A.S.M.E., called "The Code for Complete Steam Power Plants".

(2) *Object and Preparations.* Determine the object of the test, take dimensions, and note the physical condition not only of the engine but of all parts of the plant that are concerned in the determinations, examine for leakages, install testing appliances, etc., as pointed out in the general instructions.

(3) *Apparatus and Instruments.* The apparatus and instruments required for a performance test of a reciprocating steam engine are:

(a) Tanks and platform scales for weighing water (or water meters calibrated in place)
(b) Graduated scales attached to the water glasses of the boilers if the feed water is measured
(c) Pressure gages, vacuum gages, and thermometers
(d) Steam calorimeter
(e) Barometer
(f) Steam engine indicators
(g) Planimeter
(h) Tachometer, revolution-counter, or other speed-measuring device
(i) Friction brake or dynamometer if available

(4) *Methods Adopted.* Feed-Water Test. The determination of the heat and steam consumption of an engine by feed-water test requires the measurement of the various supplies of water to the boiler, that of water wasted by separators and drips on the main steam line; that of steam used for other purposes than the main engine cylinders, and that of water and steam which escape by leakage of the boiler and piping, all of these losses being deducted from the total feed water measured.

Air Pump Discharge Test. When a surface condenser is provided and the steam consumption is determined from the water discharged by the air pump, no such measurement of drips and leakage is required, but assurance must be had that all the steam passing into the cylinders finds its way into the condenser If the condenser leaks, the defect should be remedied or suitable correction should be made. The water of condensation from jackets and reheaters, if not included in the air pump discharge, should be added thereto.

Steam Meter Test. When no other method is available the steam consumption may be determined by the use of a steam meter, bearing in mind the caution that it should be calibrated under the exact conditions of use.

Steam Used by Auxiliaries. The steam consumed by steam-driven auxiliaries which are required for the operation of the engine should not be included in the total steam from which the heat consumption is calculated, but the quantity of steam thus used should be determined and reported.

(5) *Operating Conditions.* Determine what the operating conditions should be to conform to the object in view and see that they prevail throughout the trial as previously explained.

Duration of Test. A test for steam or heat consumption with a substantially constant load should be continued for such time as may be necessary to obtain a number of successive hourly records during which the results are reasonably uniform. For a test involving the measurement of feed water for this purpose, five hours duration is sufficient. Where a surface condenser is used and the measurement is that of the water discharged by the air pump, the duration may be somewhat shorter. In this case, successive half-hour records may be compared and the time correspondingly reduced.

When the load varies widely at different times of the day, the duration should be such as to cover the entire period of variation.

Starting and Stopping. The engine and appurtenances having been set to work and thoroughly heated, as previously explained, note the water levels in the boilers and feed reservoir, take the time, and consider this the starting time. Then begin the measurements and observations and carry them forward until the end of the period determined on. When this time arrives, the water levels and steam pressure should be brought as near as practicable to the same points as at the start. This being done, again note the time and consider it the stopping time of the test. If there are differences in the water levels, proper corrections are to be applied.

Where a surface condenser is used, the collection of water discharged by the air pump begins at the starting time, and the water is thereafter measured or weighed until the end of the test

Records. The general data should be recorded as previously pointed out. Half-hour readings of the instruments are sufficient, except where there are wide fluctuations. A set of indicator diagrams should be obtained at intervals of 15 or 20 minutes and oftener if the nature of the test makes it necessary Mark on each card the cylinder and the end on which it was taken, also the time of day. Record on one card of each set the readings of the steam and vacuum gages. These records should be subsequently entered on the general log, together with the areas, pressures, lengths, etc., measured from the diagrams, when these are worked up.

(6) *Calculation of Results.* The following directions are given for computing the most important results:

Dry Steam The quantity of dry steam consumed is determined by deducting the moisture, if any, found by the calorimeter test from the total amount of feed water (the latter being corrected for leakages and other losses) or from the amount of air pump discharge, as the case may be If the steam is superheated, no correction is to be made for the superheat

Heat Consumption The number of heat units consumed by the engine is found by multiplying the weight of feed water consumed, corrected for moisture in the steam, if any, and for plant leakages and other exterior losses, by the total heat of 1 pound of steam (saturated or superheated) at the pressure in the steam pipe near the throttle, less the heat in 1 pound of water at the temperature corresponding to the pressure in the exhaust pipe near the engine

Indicated Horsepower In a single double-acting cylinder the indicated horsepower is found by using the formula

$$\frac{PLAN}{33000}$$

in which P is the average mean effective pressure in pounds per square inch measured from the indicator diagram; L is the length of the stroke in feet, A is the area in square inches of the piston less one-half the area of the rod if it passes through both cylinder heads, and N is the number of single strokes per minute

Where extreme accuracy is required, the power developed by each side of the piston may be determined and the result added together

Brake Horsepower The brake horsepower is found by multiplying together the net pressure or weight in pounds on the brake arm (the gross weight minus the weight when the brake is entirely free from the pulley), the circumference of the circle whose radius is the horizontal distance in feet between the center of the shaft and the bearing point at the end of the brake arm, and the number of revolutions of the brake shaft per minute, and dividing the product by 33,000

Thermal Efficiency The thermal efficiency, that is, the proportion of the total heat consumption which is converted into work is found by dividing 2546 5 (B t u equivalent of one horsepower hour) by the number of heat units actually consumed per horsepower hour

The efficiency of the Rankine cycle is found by dividing the heat utilized per pound of steam in an ideal engine working on the Rankine cycle between the pressure and temperature in the steam pipe near the trottle and the pressure and temperatue in the exhaust pipe near the engine, by the difference between the total heat of 1 pound of steam at the throttle pressure and temperature (saturated or superheated, as the case may be), and the heat of 1 pound of water at the temperature of the steam in the exhaust pipe near the engine

The Rankine cycle ratio (or the efficiency ratio referred to the Rankine cycle) is found by dividing the efficiency of the actual engine (referred to the i h p. or br h p , as the case may be), by the efficiency of the Rankine cycle

Steam Accounted for by Indicator Diagrams at Points Near Cut-Off and Release. The steam accounted for, expressed in pounds per i.h.p. per hour, may readily be found by using the formula

$$\frac{13750}{\text{m.e.p.}}\left[(C+E)W_c-(H+E)W_h\right]$$

in which m.e.p. is mean effective pressure; C is proportion of direct stroke completed at points on expansion line near cut-off or release; E is proportion of clearance; H is proportion of return stroke incompleted at point on compression line just after exhaust closure; W_c is weight of 1 cubic foot of steam at pressure shown at cut-off or release point; and W_h is weight of 1 cubic foot of steam at pressure shown at compression point.

The points near cut-off, release, and compression referred to above are illustrated in Fig. 104.

In multiple-expansion engines the mean effective pressure to be used in the above formula is the aggregate m.e p. referred to the cylinder under consideration. In a compound engine the aggregate m.e p for the high pressure cylinder is the

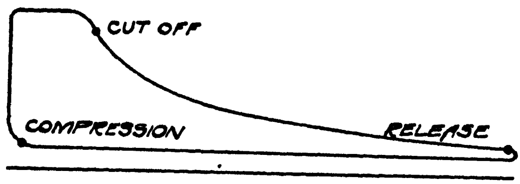


Fig 104 Indicator Diagram, Showing Points Where "Steam Accounted for by Indicator" is Computed

sum of the actual m e p of the h p cylinder and that of the l.p cylinder multiplied by the cylinder ratio. Likewise the aggregate m e p. for the l p cylinder is the sum of the actual m e p of the l p. cylinder and the m e.p of the h p. cylinder divided by the cylinder ratio.

Cut-Off and Ratio of Expansion. To find the percentage of cut-off, or what may best be termed the *commercial cut-off*, the following rule should be observed:

Through the point of maximum pressure during admission draw a line parallel to the atmospheric line Through a point on the expansion line where the cut-off is complete draw a hyperbolic curve The intersection of these two lines is the point of commercial cut-off, and the proportion of cut-off is found by dividing the length measured on the diagram up to this point by the total length.

To find the ratio of expansion, divide the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In a multiple-expansion engine, the ratio of expansion is found by dividing the volume of the l.p. cylinder, including clearance, by the volume of the h.p. cylinder at the commercial cut-off, including clearance.

Data and Results. The data and results should be reported in accordance with the form given herewith, adding lines for data not provided for or omitting those not required, as may conform to the object in view. Unless otherwise indicated, the items should be the averages of the data.

DATA AND RESULTS OF STEAM ENGINE TEST CODE OF 1915

(1) Test of .... engine located at ....

To determine ....

Test conducted by ....

Dimensions, Etc.

(2) Type of engine (simple or multiple expansion) ....

(3) Class of service (mill, marine, electric, etc.) ....

(4) Auxiliaries (steam or electric driven) ....

(a) Type and make of condenser equipment ....

(b) Rated capacity of condenser equipment ....

(c) Type of oil pump, jacket pump, and reheater pump (direct or independently driven) ....

(5) Rated power of engine ....

(a) Name of builders ....

(b) Kind of valves ....

(c) Type of governor ....

	1st Cyl	2d Cyl	3d Cyl
(6) Diameter of cylinders ....in.			
(7) Stroke of pistons ....ft.			
(a) Diameter of piston rod, each end ....in.			
(8) Clearance (average) in per cent of piston displacement			
(9) H.p. constant 1 lb. 1 rev ....h.p.			
(a) Cylinder ratio (based on net piston displacement) ....1 to			
(b) Area of interior steam surface ....sq. ft.			
(c) Area of jacketed surfaces ....sq. ft.			
(10) Capacity of generator or other apparatus consuming power of engine ....h.p.			

Date and Duration

(11) Date ....

(12) Duration .... hr.

Average Pressures and Temperatures

(13) Pressure in steam pipe near throttle, by gage ... lb. per sq. in

(14) Barometric pressure ... in of mercury.

(a) Pressure at boiler, by gage ... lb. per sq. in

(15) Pressure in 1st receiver, by gage ... lb per sq in.

(16) Pressure in 2d receiver, by gage ... lb per sq in

(17) Pressure in exhaust pipe near engine, by gage ... lb. per sq in.

(18) Vaccuum in condenser ... in of mercury.

(a) Corresponding absolute pressure ... lb per sq. in.

(19) Pressure in jackets and reheaters ... lb per sq in.

(20) Temperature of steam near throttle ... deg

(a) Temperature of saturated steam at throttle pressure ... deg

(b) Temperature of steam leaving 1st receiver, if superheated ... deg

(c) Temperature of steam leaving 2d receiver, if superheated ... deg

(21) Temperature of steam in exhaust pipe near engine ... deg

(a) Temperature of injection, or circulating, water entering condenser ... deg

(b) Temperature of injection water leaving condenser ... deg.

(c) Temperature of air in engine room ... deg.

Quality of Steam

(22) Percentage of moisture in steam near throttle or number of degrees of superheating ... per cent or deg

Total Quantities

(23) Total water fed to boilers ... lb.

(24) Total condensed steam from surface condenser (corrected for condenser leakage) ... lb.

(25) Total dry steam consumed (Item 23 or 24 less moisture in steam) ... lb

Hourly Quantities

(26) Total water fed to boilers or drawn from surface condenser per hour lb.

(27) Total dry steam consumed for all purposes per hour (Item 25 ÷ Item 12) ... lb.

(28) Steam consumed per hour for all purposes foreign to the main engine ... lb

(29) Dry steam consumed by engine per hour (Item 27 − Item 28) ... lb.

(a) Circulating water supplied to condenser per hour ... lb.

Hourly Heat Data

(30) Heat units consumed by engine per hour [Item 29 × (total heat of steam per pound at pressure of Item 13 minus heat in 1 lb. of water at temperature of Item 21)] .... B.t.u.

(a) Heat converted into work per hour .... B.t.u.

(b) Heat rejected to condenser per hour [Item 29a × (Item 21b − 21a)] (approximate) .... B.t.u.

(c) Heat rejected in form of uncondensed steam withdrawn from cylinders .... B.t.u.

(d) Heat lost by radiation .... B.t.u.

Indicator Diagrams

	1st Cyl	2d Cyl	3d Cyl
(31) Commercial cut-off in per cent of stroke .... per cent			
(32) Initial pressure above atmosphere .... lb. per sq. in.			
(33) Back pressure at lowest point above or below atmosphere .... lbs. per sq. in.			
(a) Mean back pressure above atmosphere or zero .... lb. per sq. in.			
(34) Mean effective pressure .... lb. per sq. in.			
(a) Equivalent m.e.p. referred to 1st cylinder .... lb. per sq. in.			
(b) Equivalent m.e.p. referred to 2d cylinder .... lb. per sq. in.			
(c) Equivalent m.e.p. referred to 3d cylinder .... lb. per sq. in.			
(35) Aggregate m.e.p. referred to each cylinder .... lb. per sq. in.			
(36) Steam accounted for per i.h.p. hr. at point on expansion line shortly after cut-off .... lb.			
(37) Steam accounted for per i.h.p. hr. at point on expansion line just before release .... lb.			
(a) Pressure at selected point near cut-off .... lb. per sq. in.			
(b) Pressure at selected point near release .... lb. per sq. in.			
(c) Pressure at point on compression curve shortly after exhaust closure .... lb. per sq. in.			
(d) Proportion of direct stroke completed at selected point near cut-off			

	1st Cyl.	2d Cyl.	3d Cyl.
(e) Proportion of direct stroke completed at selected point near release	....	....	....
(f) Proportion of return stroke uncompleted at selected point on compression line	....	....	....
(g) Ratio of expansion	....	....	....
(h) M.e.p. of hypothetical diagram ....lb. per sq. in.	....	....	....
(i) Diagram factor	....	....	....

Speed

(38) Revolutions per minute........ r.p.m.

(39) Piston speed per minute........ ft.

(a) Variation of speed between no load and full load .. per cent

(b) Momentary fluctuations of speed on suddenly changing from full load to half-load .... per cent

Power

(40) Indicated h.p. developed, whole engine........ i h p.

(a) I.h p. developed by 1st cylinder........ i.h.p.

(b) I.h.p. developed by 2d cylinder........ i.h p

(c) I.h.p. developed by 3d cylinder........ i.h.p.

(41) Brake h.p........ br h.p

(42) Friction of engine (Item 40 − Item 41)........ h.p.

(a) Friction expressed in percentage of i.h.p. (Item 42 ÷ Item 40 × 100)........ per cent

(b) Indicated h p. with no load at normal speed........ i.h p.

Economy Results

(43) Dry steam consumed by engine per i.h.p. per hr........ lb.

(44) Dry steam consumed by engine per br.h.p. hr........ lb

(45) Percentage of steam consumed by engine accounted for by indicator at point near cut-off........ per cent

(46) Percentage of steam consumed near release........ per cent

(47) Heat units consumed by engine per i.h.p. hr. (Item 30 ÷ Item 40)........ B.t.u.

(48) Heat units consumed by engine per br.h.p. hr. (Item 30 ÷ Item 41)........ B t u.

Efficiency Results

(49) Thermal efficiency of engine referred to i.h.p. (2546.5 ÷ Item 47) × 100........ per cent

(50) Thermal efficiency of engine referred to br.h.p. (2546.5 ÷ Item 48) × 100........ per cent

(51) Efficiency of Rankine cycle between temperatures of Items 20 and 21 ....per cent
(52) Rankine cycle ratio referred to i.h.p. (Item 49 + Item 51) ....per cent
(53) Rankine cycle ratio referred to br h.p. (Item 50 + Item 51) ....per cent

Work Done per Heat Unit

(54) Net work per B.t.u. consumed by engine (1980000 + Item 48) ....ft. lb.

Sample Diagrams

(55) Sample diagrams from each cylinder ....
(a) Steam pipe diagrams........

PRINCIPAL DATA AND RESULTS OF RECIPROCATING ENGINE TESTS

(1) Dimensions of cylinders........
(2) Date ....
(3) Duration .... hr.
(4) Pressure in steam pipe near throttle, by gage .... lb per sq. in
(5) Pressure in receivers .... lb. per sq in.
(6) Vacuum in condenser .... in. of mercury
(7) Percentage of moisture in steam near throttle or number of degrees of superheating .... per cent or deg
(8) Net steam consumed per hour .... lb
(9) Mean effective pressure in each cylinder .... lb. per sq. in
(10) Revolutions per minute .... r.p.m.
(11) Indicated horse power developed .... i.h.p
(12) Steam consumed per i h.p. hr .... lb.
(13) Steam accounted for at cut-off each cylinder .... lb
(14) Heat consumed per i.h.p. hr. .... B.t.u.

Special Tests. For an engine driving an electric generator the form of test should be enlarged to include the electrical data, embracing the average voltage, number of amperes each phase, number of watts, number of watthours, average power factor, etc.; and the economy results based on the electric output embracing the heat units and steam consumed per electric h.p.hr. and per kw.hr., together with the efficiency of the generator.

Likewise in a marine engine having a shaft dynamometer, the form of test should include the data obtained from this instrument, in which case the brake h.p. becomes the shaft h.p.

Actual Engine Test. To illustrate the application of many of the items given as obtained from the Code, a full engine test will be

taken and reported upon. This report will serve to give the order and manner in which data should be tabulated and also the method in which the report should be worked up.

DETERMINATION OF EFFICIENCY OF A BUCKEYE ENGINE UNDER DIFFERENT LOADS

Purpose

The purpose of this series of tests on the Buckeye engine located in the Engineering Laboratory of Purdue University was to determine the best efficiency under six different loads, ranging from zero to $1\frac{1}{4}$ load, by $\frac{1}{4}$ load steps, the engine running non-condensing and using 160 pounds of steam pressure, absolute.

Plan

The zero load was determined with the friction brake *I*, Fig 105, removed and the engine running free. The full load was determined by the brake load

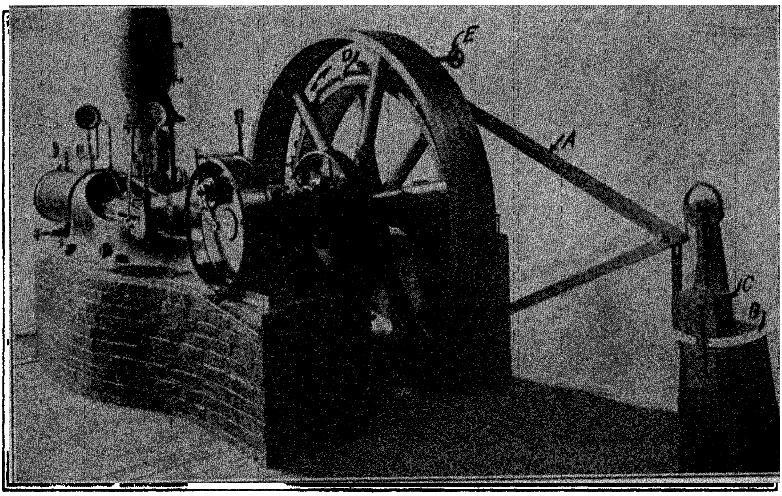


Fig 105 Buckeye Engine Fitted with Prony Brake and Indicators

which the engine carried with 25 per cent cut-off, this being the builders' rating for this type of engine. The $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, 1, and $1\frac{1}{4}$ loads were taken as 25%, 50%, 75%, 100%, and 125%, respectively, of the full load.

Steam pressure was maintained constant at the pressure indicated for the test. Each test was of one hour duration, the engine having been run under conditions of the test a length of time sufficient to permit the conditions to become constant.

Method of Conducting Test

Constant steam pressure was obtained by throttling the 5-inch steam line leading to the engine by means of the pipe line valve. This throttling action was not sufficient to cause the steam to become superheated.

The revolutions per minute were obtained by means of a revolution counter.

Indicator diagrams were taken every five minutes, 13 sets of diagrams being obtained for each hour's run.

Barometer readings were taken every 15 minutes

The amount of water was determined by condensing the exhaust steam at atmospheric pressure.

Preliminary Work

Before commencing the work the engine was placed in as good condition as was possible. The governor was adjusted in order to reduce friction; play was taken up in the valve gear and the valves were carefully set to give equal cut-off on both ends at full load; all stuffing boxes were repacked, the brake wheel was turned up and brake recalibrated

The pressure in the engine supply line was obtained by tapping a ¼-inch pipe into the main, about 3 feet from the valve. This ¼-inch pipe was connected to a large steam gauge which faced the operator of the throttling valve, thus enabling him to watch the gauge all the time and maintain a constant pressure.

Observed Data

In each test the following observations were taken:

Steam pressure, constant throughout
Brake load
Revolutions per minute
Weight of condensed steam
Barometer
Indicator diagrams

Results

Having the above data it becomes possible to calculate the following:

(1) Per cent of cut-off, head end and crank end.
(2) Mean effective pressure (m.e p.), head end and crank end.
(3) Indicated horsepower, head end and crank end and total.
(4) Brake horsepower (b.h.p.).
(5) Friction horsepower (f.h.p.).
(6) Mechanical efficiency.
(7) Pounds steam, per indicated horsepower per hour and per brake horsepower per hour.
(8) British Thermal Units per hour, per indicated horsepower and brake horsepower per hour.
(9) Thermal efficiency.

Constants and Formulas. The constants of the engine and formulas employed in obtaining the calculated items in the summary of results, are as follows:

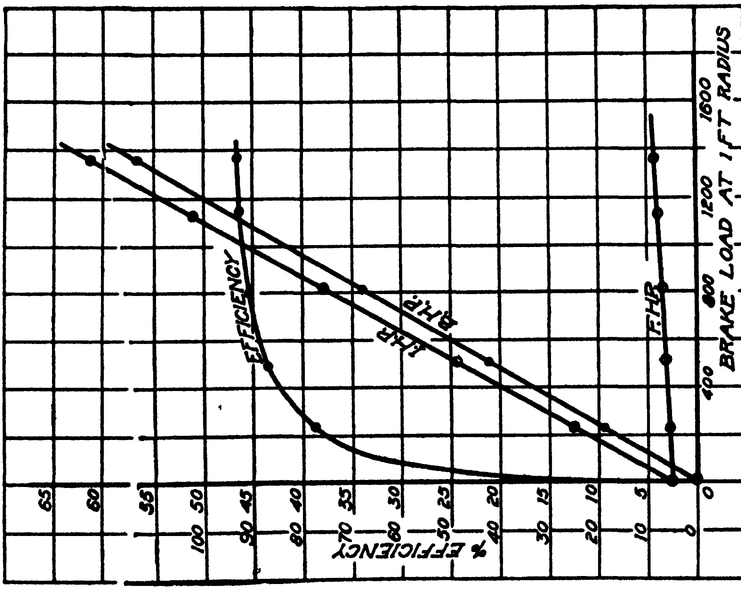


Fig 106. Horsepower and Efficiency Curves for Buckeye Engine

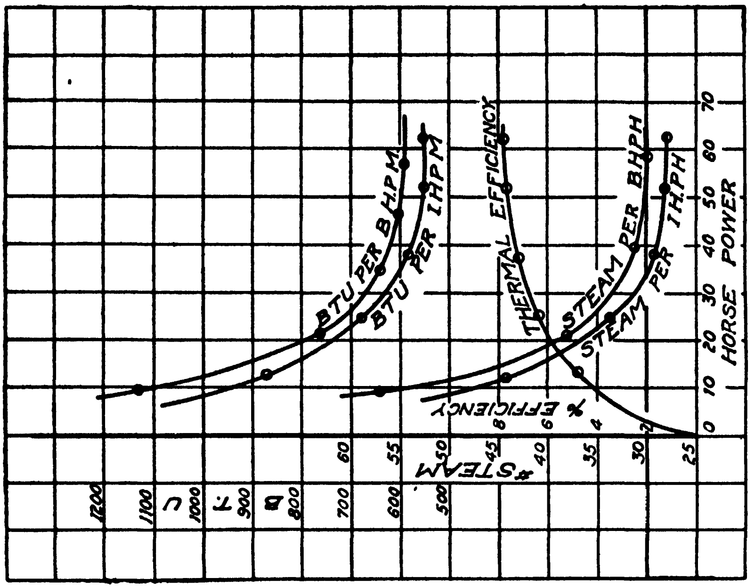


Fig 107. Steam Consumption B T U and Thermal Efficiency Curves for Buckeye Engine

Diameter of cylinder, 7.75 inches.
Piston rod diameter, 1.437 inches.
H.E. area, 47 173 square inches; c.e. area, 45.55 square inches.
Radius of brake arm, 38.25 inches, equals 3 185 feet
Clearance, head end 6.15%; crank end, 6.765%.
Normal speed, 220 revolutions per minute
Heat value of 1 horsepower, 42.42 British Thermal Units.
Heat value of 1 pound of steam, above 32°F. for 160 pounds absolute. 1,192.8 British Thermal Units.
Gauge pressure 15 pounds (approximate) less than absolute pressure.

The horsepower constants are as follows:

H.E.—i h.p. Constant = .001787. (See "Steam Engine Indicators.")
C.E.—i.h.p. Constant = .001726.
B.H.P. Constant = .00060695.

Item (3). At observed revolutions per minute (r.p.m.):

H.E.—i.h.p. = .001787 × h.e. m.e.p. × r.p.m.
C.E.—i.h.p. = .001726 × c.e. m.e.p. × r.p.m
Total i.h p. = h e. i.h.p. + c.e. i.h.p.

It often happens that the engine is not operated at the desired speed just at the instant of taking the reading, hence a correction must be made if the indicated horsepower is to be expressed and recorded for the normal speed. Therefore i.h.p. = total i.h p. × 220 ÷ observed r p m.

Item (4). At observed r.p.m. determined as follows:

B.H.P. = .00060695 × pounds brake load × r.p.m.
B.H.P. = .0001904 pounds brake load at 1 foot radius × r.p.m.
At 220 r.p.m., corrected b.h.p. = b.h.p. × 220 ÷ observed r.p.m.

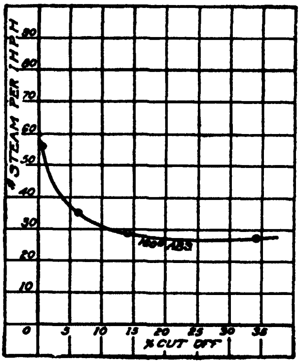


Fig. 108. Steam Consumption for Different Cut-Offs

Item (5). At 220 r.p.m., the f.h.p. = total i.h.p − b.h.p.

Item (6). At 220 r.p.m., the mechanical efficiency = b.h.p. ÷ total i.h.p.

The pounds of steam per hour at 220 r p.m = pounds of steam per hour at observed r p.m. × 220 ÷ observed r.p.m. The B.T U. supplied per hour = corrected pounds of steam per hour × total British Thermal Units in 1 pound steam, at given absolute pressure above 32°F.

Item (7). The pounds steam per i.h.p. per hour = corrected pounds steam per hour divided by corrected i.h.p.

The pounds of steam per b.h.p. per hour = corrected pounds steam per hour divided by corrected b.h.p.

Item (8). The British Thermal Units per i h.p per hour = total British Thermal Units supplied divided by 160 × corrected i.h.p.

The British Thermal Units per b.h.p., per hour = total British Thermal Units supplied divided by corrected b.h.p. × 160.

Item (9). The Thermal Efficiency = 42.42 British Thermal Units divided by British Thermal Units per i.h.p. per hour

TABLE VII

Indicator Diagram Data for Buckeye Engine Test

Steam Pressure 160 # Abs.							
Cyl. End	Card No.	No Load		1/4 Load		1/2 Load	
		%C.O.	M.E.P.	%C.O.	M.E.P.	%C.O.	M.E.P.
Head End	1		.000	1.58	14.73	5.00	28.41
	2		.785	1.32	13.98	5.52	28.17
	3		.773	1.05	14.14	4.98	28.37
	4		.259	1.05	14.73	4.71	26.70
	5		1.040	1.05	13.65	5.00	26.82
	6		.000	1.05	14.45	4.46	26.78
	7		.000	.79	14.13	4.47	26 30
	8		.521	.79	13.41	4.71	26.18
	9		.675	1.05	12.90	4.71	27.22
	10		1.295	1 05	12.63	4.97	27.22
	11		.675	1.05	12.90	4.71	27 22
	12		.529	1.06	14.80	4.73	26.30
	13		.779	1.06	12 44	4 97	27.22
	Av.		.5638	1.073	13757	4.841	27.14
Crank End	1		4.325	77	18.58	7.41	36.59
	2		4.325	.76	1850	7.66	36 80
	3		3.805	.77	18.62	7.65	36 46
	4		3.560	.77	18.58	7.65	36.20
	5		4.055	.77	19.13	7.65	36.20
	6		3.567	.77	18.92	7.66	36.30
	7		4.340	.77	18.43	7.65	36 20
	8		3.805	77	18.37	7.41	36.30
	9		3.805	76	17.70	7.64	37.63
	10		3.785	.76	17.95	7.41	37.81
	11		3.805	.76	17.95	7.65	36.75
	12		3.560	76	17.75	7 40	36.20
	13		4.555	.76	17.95	7.69	37.18
	Av.		3.9445	.7653	18.338	7578	36.655

TABLE VIII

Indicator Diagram Data for Buckeye Engine Test

STEAM PRESSURE 160 # ABS.							
CYL. END	CARD NO.	¾ LOAD		FULL LOAD		1¼ LOAD	
		% C O.	M.E.P.	% C O.	M.E.P.	% C O.	M.E.P.
HEAD END	1	13.85	46.70	24.90	63.25	33.92	77.60
	2	13.85	44.95	24.05	63.15	35.22	77.60
	3	13.69	44 75	24.85	62.85	35.32	77 80
	4	13.88	44.75	25.15	63.35	34.90	77.00
	5	14.40	44 50	24.52	64 65	35.41	76.60
	6	14.21	45.50	25 15	64.15	35 26	77.10
	7	13.85	44.65	24.21	64.20	34.27	77.55
	8	13.57	45.15	25.05	63 76	34.87	77 05
	9	13 65	44.90	25.15	64.90	36.21	78.95
	10	14.15	46.20	24 35	63.35	34.05	77.50
	11	14 70	45.40	24 42	63.25	34.07	77 95
	12	13.92	44.40	25 15	64 15	34.80	78 05
	13	14.11	44.15	25.20	64 50	34.72	77 95
	AV	13.98	45.073	24.78	63.80	34 84	77.537
CRANK END	1	14.05	53.15	24.80	67.85	31.87	82.05
	2	14.50	52.80	24.75	67.30	34.25	82.30
	3	14.05	52.15	25.10	68.70	33 59	81.80
	4	14.32	53.45	24.05	67.50	32 59	80.80
	5	14.54	51.80	24.67	67.50	33 32	81.31
	6	14.11	53.10	24.80	68.35	32.83	81.30
	7	14.22	52.55	24.28	68.40	32 83	80.80
	8	14.54	52.25	24.61	68.00	32.83	81.00
	9	14.28	51.80	24.80	68.80	34.25	81.80
	10	14.50	51.85	24.38	69.06	32.90	80.25
	11	14.54	52.25	24.60	68.65	33.07	80.75
	12	14.54	53.30	24.58	68.55	33.52	80.00
	13	14.76	52.15	24.90	68.50	32.63	80 20
	AV.	14.38	52.527	24.59	68.30	33.11	81.10

TABLE IX

PERFORMANCE OF
UNDER DIFFERENT CUT-OFF'S.
SUMMARY

OBSERVED						AV. M.E.P.		I. H. P.			
LOAD	#BRAKE LOAD AT 1 FOOT RADIUS	R.P.M.	BAROMETER INS. OF MERCURY	#STEAM CONDENSED PER HOUR	PER CENT CUT-OFF	HEAD END	CRANK END	HEAD END	CRANK END	TOTAL AT OBSERVED R.P.M.	TOTAL CORRECTED TO 220 R.P.M.
0	0	2871	2982	4345		564	3945	289	1.953	2.242	172
¼	227.5	225.5	2983	,550.0	.97	1376	18.39	545	7.13	12.66	1235
½	5310	2229	2981	8490	621	2714	3666	1082	14.10	2492	24.58
¾	8150	2219	29.80	1096.0	14.2	4507	5253	1786	2010	37.96	37.62
1	11260	217.4	2978	1401.0	246	6380	6774	2480	2562	5042	51.00
1¼	13630	2097	2979	1614.5	34.0	7754	8110	29.00	2930	58.30	61.20

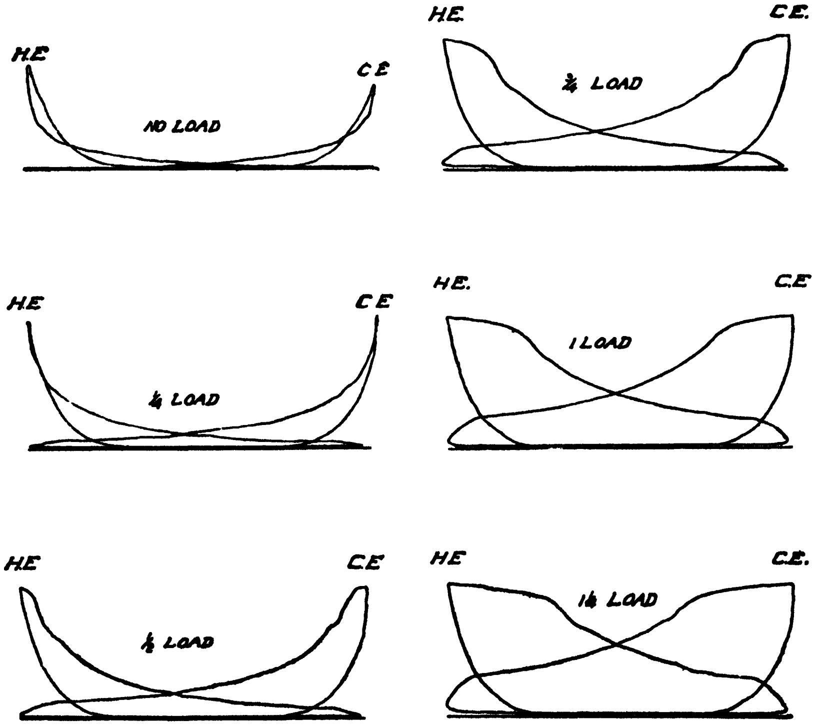


Fig. 109. Indicator Diagrams Taken During Test of Buckeye Engine

TABLE IX—Continued

THE BUCKEYE ENGINE
STEAM PRESSURES 160# ABS.
OF RESULTS.

CALCULATED										
B.H.P.										
AT OBSERVED R.P.M.	CORRECTED TO 220 R.P.M.	F.H.P. AT 220 R.P.M.	MECHANICAL EFFICIENCY	#STEAM PER HR. CORRECTED TO 220 R.P.M.	TOTAL B.T.U. SUPPLIED PER HOUR	#STEAM PER I.H.P.H.	#STEAM PER B.H.P.H.	B.T.U. PER I.H.P.M.	B.T.U. PER B.H.P.M.	THERMAL EFFICIENCY
0	0	2.35	0	332.5	396600	141.4		2810		0
9.75	9.54	2.71	77.2	547.0	652500	44.25	57.4	879	1140	4.82
21.81	21.51	3.07	87.5	837.0	998350	34.1	38.95	678	774	6.29
34.41	34.18	3.44	90.8	1085.0	1294300	28.85	31.80	573	632	7.25
46.55	47.17	3.88	92.5	1418.0	1691500	27.8	30.5	552.5	606	7.68
54.45	57.07	4.13	93.3	1695.0	2022000	27.75	29.7	551	590	7.70

Plotted Results. On curve sheet shown as Fig. 106, are plotted to pounds brake load at 1′ radius, the i h p , b h p , f.h p , and mechanical efficiency.

On curve sheet shown as Fig. 107, are plotted to horsepower the pounds steam per i h p per hour, the pounds steam per b h p. per hour, the British Thermal Unit per i h p per minute, the British Thermal Unit per b.h p. per minute, and the thermal efficiency.

On curve sheet shown as Fig. 108, is plotted a curve which shows the steam consumption for the different per cents of cut-off.

Conclusions and Comparisons

An examination of the curves shows a marked increase in economy of the ½ loaa over the ¼ load; a smaller increase in economy of the ¾ load over the ½ load, and a still smaller increase in economy of the full load over the ¾ load; but the full load and 1¼ load have the same steam consumption per i.h.p. per hour indicating that the engine is operating most economically throughout this range.

The tests indicate a very good range of economical operation from ¾ load to 1¼ load, and although the steam consumption is higher than the best recorded results for other engines of greater horsepower, yet the results obtained are very good considering the relatively small size of the engine.

Appendix

The engine under test was a 7¾″×15″ type "B" Buckeye engine which had been rebuilt from the old type of flat valve to a piston valve engine. The following information was supplied by the Buckeye Engine Company:

Lap $\frac{11}{16}$″, Lead $\frac{1}{16}$″, Compression 2¼″, Exhaust laps, ⅛ and $\frac{1}{16}$″, Clearance 5.6%, Cut-off 25%. Weight of reciprocating parts 150 pounds.

The arrangement of the brake apparatus may be seen in Fig. 105, in which *A* is brake lever, *B* is calibrated brake load arc, *C* is brake pendulum, and *D* is brake wheel.

Cooling water for the brake enters through a hose not shown in the illustration. The direction of rotation of the brake wheel is indicated by the arrow near *D*. By means of the hand wheel *E*, the brake load is applied and regulated The brake was carefully calibrated before beginning the test.

Computation of Constants

H E piston displacement $= \frac{47.173 \times 15}{144 \times 12} = 41$ cu ft.

C E piston displacement $= \frac{45\ 55 \times 15}{144 \times 12} = 396$ cu. ft.

H E clearance $= \frac{.0252}{41} = 6\ 15\%$

C E. clearance $= \frac{0268}{.396} = 6.765\%$

H E —i h p Constant $= \frac{15 \times 47.173}{12 \times 33000} = 001787$

C E —i h.p Constant $= \frac{15 \times 45\ 55}{12 \times 33000} = 001726$

B H P Constant $= .00060695$

Thermal Efficiency Constant $= \frac{33000}{778} = 42\ 42$

Tables VII and VIII contain information from the indicator diagrams, and Table IX is a general summary of the observed and calculated results of the tests Fig 109 shows sample indicator diagrams taken during the test

STEAM ENGINE TROUBLES AND REMEDIES

Manufacturers supplying reciprocating steam engines for commercial uses usually make it a part of their business to see that the machines are properly installed with suitable foundations, steam and exhaust piping, and accessories. They usually furnish a competent man to be present when the engine is first placed into commission to see if it operates satisfactorily. So at the beginning we usually find stationary reciprocating steam engines operating quietly and in a manner acceptable to all concerned. As time goes on, however, and a number of different operators have had charge of the plant, and natural wear of the various parts occur, operating trouble will be experienced. Then again, accidents are liable to occur from time to time which will require immediate attention from those in charge in order that the machines may be kept running. These conditions make it imperative that the operator in charge be prepared, as far as possible, to make the necessary repairs when the emergency arises.

During the life of an engine which is put to continuous duty in a power plant or a manufacturing plant, any one of a large number of troubles or accidents may occur. Only a limited number of cases can be discussed.

Broken Cylinder Casting, Cylinder Head, or Piston. *Cause of Water in Cylinder.* An accident due to water in the cylinder usually occurs at a time when the machine is being started after having been shut down for a considerable period, in other words, when the piping and parts have had sufficient time to cool down. It is assumed that a steam separator is located in the steam line at a point just ahead of the throttle, if the engine is located any great distance from the boiler, and that cylinder drain cocks are provided if the engine is of the slide or piston valve type. The accident usually happens because of the carelessness of the operator. Sufficient time must be allowed to get the cylinder and parts warmed and all entrained water worked through the cylinder, before bringing the engine up to full speed. It sometimes happens that if the water in the boiler is being carried at too high a level and is dirty, water will be carried over into the cylinder in sufficient quantities to cause an accident. Some types of engines have pressure relief valves attached to the ends of the cylinder which will open when any undue pressure occurs within the cylinder.

Breakage of Cylinder Parts. Such accidents may result in a breakage of the cylinder head, cylinder casting, piston, or all three. Cases are on record where the machine was a complete wreck, rupturing, in addition to the parts mentioned, other parts, such as the crank pin, crank disc, connecting rod, piston rod, etc. Usually if but one part is broken, an additional one may be secured from the makers and be placed into position by the operator. If, however, the damage is very serious it is better to let the builder handle the matter of repairs as seems best.

Knocking or Pounding. *Difficulty of Locating Noise.* The operator in charge of a smoothly running engine takes great pride in showing it to visitors, but when a click or pound is heard he naturally feels that these defects reflect upon his skill, and, whether alone or in the presence of visitors, it is a constant source of annoyance. He will put in many hours of overtime for

which he makes no charge in an effort to locate the cause of these noises, which to outsiders are of no consequence but to his ear become almost unbearable at times. When the engine is working under a light load the engine runs so smoothly that he fancies the trouble has been reached, but later when it is operating under normal load the exasperating pounding begins, and, to aggravate the case, the bearings that formerly had given no trouble begin to heat from being too closely adjusted.

There are few people outside of the engineering profession who really know how perplexing a knocking or pounding in an engine may become. In the first place it is frequently very difficult to locate. Go to the cylinder and it seems to be there. Stand at the crank and the noise is there. In such cases the sensitive ear of the earnest and sincere operating engineer can hear nothing else, and its continuance affects both his mind and body in a way that it is difficult to explain. Many such cases require much thought and patience to eventually locate the trouble.

Causes of Pounding and Knocking. Improper Alignment. Improper alignment is perhaps one of the most common causes of pounding, and this is a defect which every operator should be able to correct by adjusting the working parts to a line. The alignment may have been correct when the machine was first installed, but continued use has brought about changes which make realignment necessary.

Uneven Wear of Bearings. There are many other causes of pounding to which attention might be called, as the obscurity of some of these might cause them to be overlooked. For instance, wrist pins, crosshead pins, and crank pins naturally wear unevenly and eventually will not be circular in section. In such cases, if the bearings are properly adjusted at the dead center points they probably will be quite loose at the quarter points. If properly adjusted at the quarter points they will be too tight at the dead centers and will give trouble by heating. The only remedy in these instances is either to replace the pins or have them trued up in an engine lathe and the bearing readjusted to the new size.

Loose Flywheel or Belt Pulley. Perhaps the most difficult pound or knock to locate is that caused by a loose flywheel or

belt pulley. When such conditions are found, the remedy to apply is at once evident.

Worn Shaft Governor Parts. Engines fitted with shaft governors, which usually have several joints and movable parts, frequently become noisy due to these parts becoming worn. To remedy this trouble, it is usually necessary to rebore the holes in the various parts and make new pins to fit. It may be necessary in certain cases to bush some of the holes. It is not uncommon to hear pounding from such engines caused by the governor fluctuating, the parts not remaining relatively at rest for the load being carried. Such pounding is usually traceable to one of two causes—either the governor ports themselves are set up too snug or the packing gland on the valve rod is turned up too tight.

Causes of Noises in Cylinder and Steam Chest. Occasionally one hears an engine operating with clicking noises emanating from the cylinder and steam chest. These noises may be caused by a number of different things. If the piston packing ring overtravels the steam port too great a distance, the steam pressure may compress the ring momentarily, which will assume its normal shape when the piston moves ahead or back, as the case may be, thus causing a "spring slam" at each end of the stroke. Improper valve setting may be the cause of such a knock or click as above mentioned. Should the compression pressure exceed the initial steam pressure, the valve will be lifted from its seat. The simplest way of correcting this trouble would be to take indicator cards and reset the valve or valves to give the proper compression. If the engine has been in service a number of years the piston packing rings may become sufficiently worn to cause a rattle or clicking noise. In such cases new rings should be fitted to the piston. In engines where the piston is made up of several parts, wear on these parts would eventually result in producing noises.

If the valve setting has become deranged through carelessness or otherwise to such and extent that the power of the engine is unequally divided between the two ends of the cylinder, the effect may be to produce a pound. This can readily be remedied by the aid of a steam engine indicator in readjusting the steam distribution. Adjustments should be made until the power is

equally divided between the two ends. This principle applies to compound engines as well as simple engines.

Improper Adjustment of Main Crank-Shaft Bearing. The main crank-shaft bearing is quite often the cause of pounds or knocks. This part is made up in so many different forms that any specific directions seem impracticable. Suffice to say, that the pound is due to improper adjustment and that if the parts are not worn too greatly the correct adjustment should not be a difficult matter.

Broken Flywheel. *Effect of Sudden Changes in Speed.* Technical literature is profuse with records of disasters caused by the failure of flywheels in use on engines when in service. In many instances the destruction is so complete that no evidence remains as to the cause of the failure. Perhaps the most common cause of flywheel failure is due to some fault in the operating condition of the governor. On this account it is extremely important that the operating engineer give careful attention to the working condition of all parts of his engine and especially the governing system. It is true that in the casting of the flywheel internal stresses may be occassioned due to unequal expansion and contraction which might in time cause failure, but the wheel is always designed with a large factor of safety to take care of such possible conditions. So it seems reasonable to assume that usually flywheel failures are caused by too sudden changes in speed. This sudden change in speed may be caused, as previously mentioned, by a defect in the governing mechanism, a slug of water coming into the cylinder through the steam pipe, or by a failure of some part of the engine mechanism, which would tend to lock the movement of the parts, such as a broken piston or a piston rod becoming loosened sufficiently to permit the piston to strike one end of the cylinder.

Effect of Worn Valve Packing Rings. Cases are on record where engines of the piston valve type have been in service for a number of years and the valve packing rings have become worn with the result that the governor would not control the speed at light loads. This was due to the fact that the worn condition of the valve permitted enough steam to leak into the cylinder to cause the speed of the machine to greatly increase. In all such engines, it is desirable to keep the valve packing rings in perfect repair not

only as a safeguard against accidents caused by overspeeding but also as an aid in maintaining the steam economy of the plant.

Maintaining Steam Economy. One of the important points to which an operating engineer should give constant attention is the steam economy of the engines placed under his charge. Information on this point can be secured from time to time by economy tests. Maintaining maximum steam economy of the plant is not only of inestimable value to the engineer himself but is dollars and cents to his employer. The steam consumption of an engine should improve up to a certain time as the piston and valve or valves become worn to a more perfect bearing. Later, however, as the cylinder, piston and rings, valve or valves, etc., become worn, they cease to remain steam tight and, as a result, permit steam to leak through without doing any real work. For this reason the engineer should keep himself informed as to the condition of his engines. If the cylinder is worn "out-of-round", it should be turned over to a reputable machine shop or returned to the maker for reboring. If the cylinder is in good condition but a piston ring is badly worn or broken, thus permitting steam to leak through, the broken ring should at once be replaced by a new one. It is preferable to secure the new ring from the builder if possible, since the builder has the proper machines and tools for turning out perfect rings. The ordinary shop, even with its best mechanics, if they have had little or no experience in such work, is very liable to turn out an imperfect job. It is very likely that an imperfectly made ring replacing a broken or worn one would show little or no improvement in steam economy.

Enlarged Vacuum Pump Valves. In plants using surface condensers, we frequently find the direct-acting type of wet vacuum or air pumps installed. These pumps handle the condensate from the condensers as well as exhausting the air from the system to a greater or less extent. With such equipment it is possible to maintain a vacuum of about 26 inches. This type of pump is usually fitted with composition rubber valves which eventually become enlarged and distorted owing to the action of the oil in the steam used for cylinder and valve lubrication. As a result of this condition, it will be found that eventually the valves do not function properly and that the vacuum will be materially

decreased, thus decreasing the power capacity of the plant and materially increasing the steam consumption. These conditions make it essential that the air end of the pump should receive frequent inspections.

Piston Rod and Valve Rod Packing Troubles. The piston and valve rods are made steam tight by one of two methods, namely, by the use of a stuffing box, where some form of packing is placed around the rod and securely held in place by means of a gland, or by the use of some form of metallic packing. Either form, if properly constructed and adjusted, will perform its function very satisfactorily. The metallic packing frequently will permit steam to blow through when the engine is started after a shutdown, but this steam blow usually ceases after the parts become thoroughly heated. When a steam blow occurs where a stuffing box is used, the operator will usually apply a wrench and tighten the gland until the blow stops. In such cases the wrench should be used very cautiously. Many times the rod will become overheated due to the gland being turned up too tight. This usually occurs when the packing has been in use a considerable length of time and has become very hard. When this condition exists the packing should be replaced. The safest policy to follow after repacking a stuffing box is to tighten the gland only as much as is necessary to prevent a steam blow. It will need watching for a time and occasional tightening.

Superheating and Lubrication. The use of superheated steam in power plant operation has in every instance demonstrated economic advantages. One of the things which operated to discourage the use of superheated steam at the start was the trouble experienced in lubrication. In almost every instance the trouble arose because of the fact that the same oil was being used as when saturated steam was being generated. It has been shown many times in practice that a grade of oil sufficiently high for saturated steam service may not prove satisfactory when used in installations where superheated steam is used. This fact is especially emphasized if a high superheat of, say, 200 degrees or more is employed. In such cases a high-grade oil is recommended. All reputable oil refineries now refine lubricating oils which they guarantee for such classes of service.

Lining an Engine. In the erection of a new engine, if it be of considerable size, the manufacturers usually send one of their experienced erecting engineers to superintend the work in order to insure satisfactory operation. In established plants, where the engines have become worn and where settlement has taken place in the foundations, the realignment of one or more of the machines may become necessary. Where such instances arise the operating engineer is usually given the responsibility of superintending the work. The following suggestions may be of some service to the inexperienced. The presentation must of necessity be abbreviated.

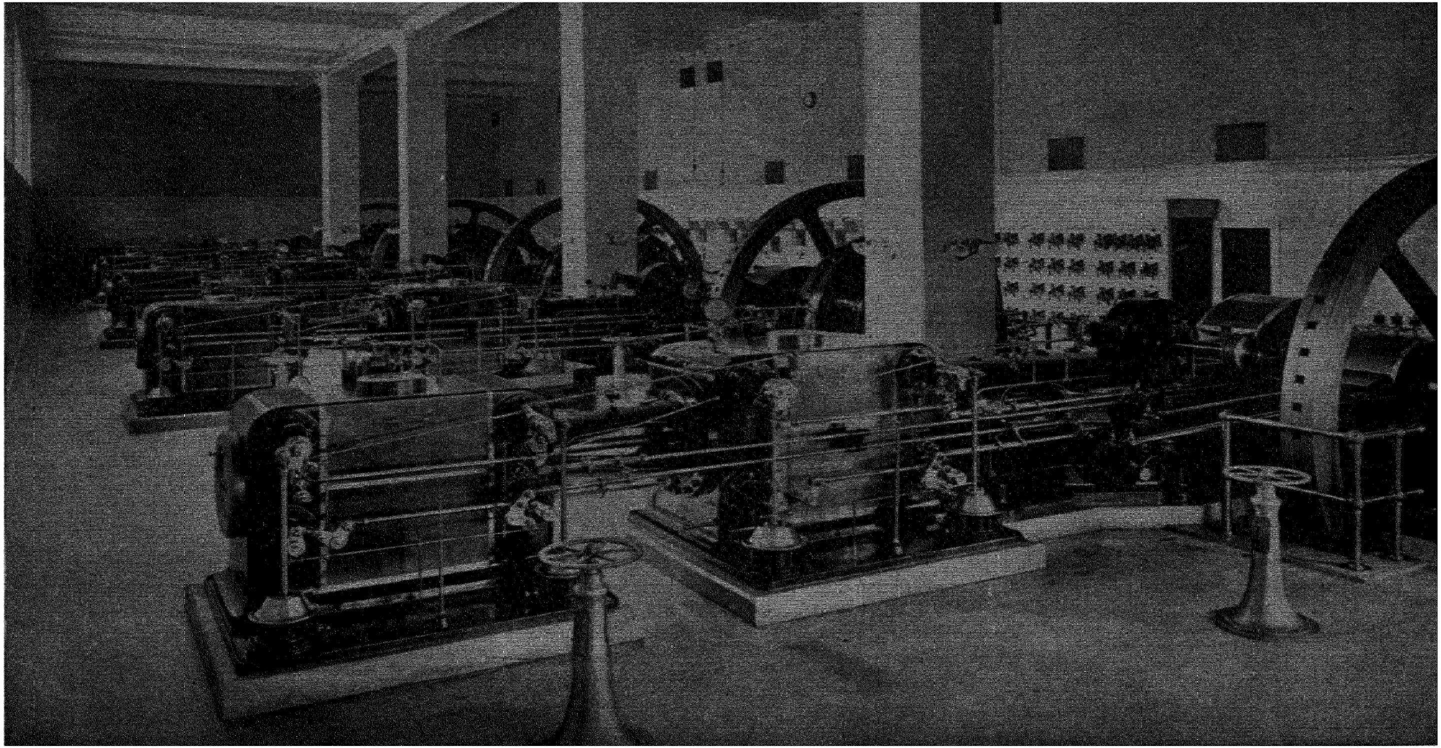
Building Foundation. To begin with, if a foundation is to be built, it should be made in accordance with the design of the engine builders. Its size will be governed largely by the type of engine and by the character of the soil in the particular locality. A substantial wood templet should first be prepared which would show the correct location of the foundation bolt holes in the engine base. This templet should be used in the construction of the foundation to properly locate the different positions of the foundation bolts, which should be embedded in the foundation to such a depth as would insure their security. For a portion of the distance down from the top of the foundation, the bolts should be surrounded by iron pipe of suitable dimensions to permit a small amount of lateral movement. This will permit correction for any slight errors of measurement in locating the bolt holes. The top of the foundation should be made level. When ready to receive the engine bed, it should be carefully placed in position and made level by the use of at least two spirit levels, holding the bed in the proper position by means of small wedges. Cement grout should then be poured under the bed and permitted to firmly set, when the small wedges can be removed and the holes filled.

Locating Center Line. It is assumed that the engine has already been dismantled, that is, the valve gear, connecting rod, cylinder head, piston and rod, and crosshead have all been removed. In securing the proper alignment the first step is to take a slotted stick or piece of metal and secure it across the head end of the cylinder by means of the cylinder head stud bolts. Draw a fine linen line or fine wire over this stick and through the

center of the cylinder out between the guides and attach it to an upright stick at the crank end of the bedplate, nailed to the floor or clamped to the bedplate. This line should then, by careful calipering, be made to pass directly through the center of the counterbore at each end of the cylinder. The measurements are made from the counterbore because this portion of the cylinder has been subjected to the least amount of wear. It is not advisable to make measurements to locate the center line in the stuffing box at the crank end of the cylinder. Sufficient time should be taken and great care exercised that this line is accurately located as much depends upon the accuracy of this first work. After the line is finally located it is well to examine the guides and determine their parallelism.

Placing Engine Shaft in Position. The next step is to lower the main engine shaft into the bearings, the outboard bearing being first loosely located. Now turn the crank first one way then the other, shifting the outer end one way or the other, until that part of the crank pin that is to be in line with the center line of the connecting rod shall be exactly over the line; also, so that the turned surfaces, if turned, of the crank shall be parallel to the line. Next make a half-turn of the shaft as nearly as possible and see that the pin maintains the same relative position to the line. Finally, carefully place the shaft in a horizontal position by the aid of spirit levels and the plumb line. It is well in checking this last work to run a second line at right angles to the center line from which to make measurements. When the engine shaft is finally located as directed, fasten the outboard bearing block securely and attach the bearing caps in place.

Checking Work. Before removing the carefully located lines, it is well to go over the work from the very beginning to check the adjustments. When this has been completed the lines may be removed and the work said to be correct and the process of assembling the parts begun.



FIVE TANDEM-COMPOUND NON-CONDENSING RICE AND SARGENT CORLISS ENGINES OPERATING DIRECT-CURRENT GENERATORS IN PARALLEL

Courtesy of Providence Engineering Works, Providence, Rhode Island

STEAM ENGINE INDICATORS

INTRODUCTION

The steam engine indicator is an instrument designed to make an accurate graphical diagram of the pressure of the steam in the engine cylinder at all points of the stroke. This diagram affords a means for studying the performance of the steam engine.

The indicator serves two very important purposes, although many other results are obtained by its use. (1) *In the hands of an experienced engineer, it enables him to discover any defects in the design or setting of the valve mechanism.* (2) *It also indicates whether the steam ports are large enough and, in fact, a proper interpretation will disclose the exact condition of the design and operation.* Thus the engineer can determine whether any change in the operation of the moving parts is advisable.

The information that may be obtained by an intelligent use of the indicator is of very great value to the engineer. The power of the engine at any time and under any condition may be determined; many facts can be accurately obtained that could not be secured in any other way; many things about the steam engine that before seemed mysterious are now made clear. Its value is so universally recognized that almost all builders of steam engines apply indicators to their engines and adjust the valves and moving parts before sending the engine away from their factories. For these and other reasons which might be mentioned, it is seen that the indicator has played no small part in the development of the steam engine.

In the early development of the steam engine by James Watt, he realized that some means should be provided whereby the internal action of the steam, valves, etc., could be watched or their behavior interpreted. As a result of this apparent need, the indicator was devised, the first forms being crude in their construction, but the underlying principles being the same as are found today in the modern instrument. It is, therefore, of interest to note that the changes

made in the indicator since its advent have been largely in constructional details rather than in principle. The moving parts of the earlier indicators were exceedingly heavy; on this account, the inertia of the moving parts often distorted the indicator diagram to such a degree that the results obtained were unreliable. The older types would give fairly accurate results on slow-speed engines but were useless on high-speed engines on account of the comparatively great weight of the pencil mechanism and other moving parts.

The modern indicator is almost perfect in construction. All of its parts are as light as good design will permit and it is conveniently manipulated and easily adjusted. It may and does at times, however, record pressures incorrectly. Some of the most common errors, which are often misleading, will be discussed later.

In order to have an intelligent understanding of the use and care of an indicator, it is necessary to become familiar with its construction, and to that end, a description of three well-known makes will be given, viz, the Crosby, Tabor, and Thompson.

TYPES

It will be observed that indicators do not differ in detail very materially, their chief difference being found in the pencil mechanism. In order to make a discussion of the construction logical in development, it is well to note first an improved form of the Watt indicator.

Watt Indicator. The Watt indicator, Fig. 1, consists of a steam cylinder *S*, about 1 inch in diameter and 6 inches long, in which a solid piston *P* is accurately fitted. A spiral spring *A* is attached to this piston, and controls the motion of a pencil *a*, which is also attached to the piston. This pencil can operate on a sheet of paper fastened to a sliding board *B*. This board moves back and forth by means of a weight at one end and a cord at the other which is connected to some reciprocating part of the engine. The indicator cylinder *S* may be put in communication with the engine cylinder by means of the cock *C*. With this instrument, a complete diagram can be taken. When cylinder *S* is put in communication with the engine cylinder by means of the cock *C*, pencil *a* is raised or lowered precisely as the intensity of the pressure in the cylinder varies. This variation of height of pencil or pressures is registered upon a card

carried by the board *B*. As the board *B* is moved in exact coincidence with the piston of the engine, by being connected to some reciprocating part, the resulting card gives an exact indication of the pressure in the cylinder for all points of the stroke. The vertical dimensions of the card, commonly called the *ordinates*, indicate the pressures; the horizontal dimensions, or *abscissas*, indicate the simultaneous positions of the piston.

It is a natural transition from the earlier form shown in Fig. 1 to the modern indicator, as the intervening changes have been largely in the perfection of the recording mechanism and in the refinement of details, as will be pointed out later.

Crosby Indicator. The Crosby indicator is illustrated in cross-section in Fig. 2. The indicator cylinder *4* is connected to the steam engine cylinder by means of the loose nut *7*. The steam passes from the steam engine cylinder to the indicator cylinder through the passage *6* and acts on piston *8*. The indicator cylinder is very carefully designed and constructed so that the piston will have perfect freedom of movement for various pressures. The annular cavity between *4* and *5* serves as a steam jacket and permits *4* to expand and contract freely.

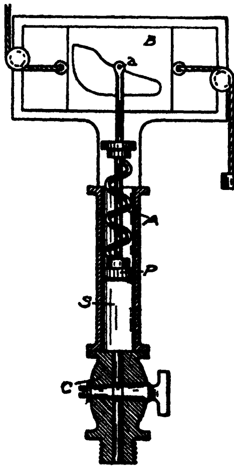


Fig 1 Original Watt Indicator

Piston *8* is made of a good quality of steel and is hardened to prevent its surface from wearing. It is ½ square inch in area. Small grooves around its outer surface provide a steam packing, and the moisture and oil which collect in these grooves prevent too much leakage of steam past the piston. At the center of the piston is a boss or hub which projects both upward and downward. The upper part of the hub is threaded inside to receive piston rod *10*. The upper edge of this hub is so formed that it fits nicely into a circular groove in the bottom side of the nut of the piston rod. The hub also has a slot cut diametrically across it, which permits the flat portion of the spring with head to fit on a curved bearing on the piston screw *9*. When making connection between piston *8* and piston rod *10*, it is very essential that the hub shank fit tightly against the bottom of

the circular groove in the bottom of the shoulder of the piston rod. If this connection is correctly made, a perfect alignment of the piston is assured.

The swivel head *11* is threaded at the bottom, so that it can be screwed into the piston rod. By so doing the height of the pencil and, therefore, the atmospheric line can be raised or lowered as desired.

Cap *2* is an important part of the indicator as it holds all the moving parts in place and guides the piston. It has two projections

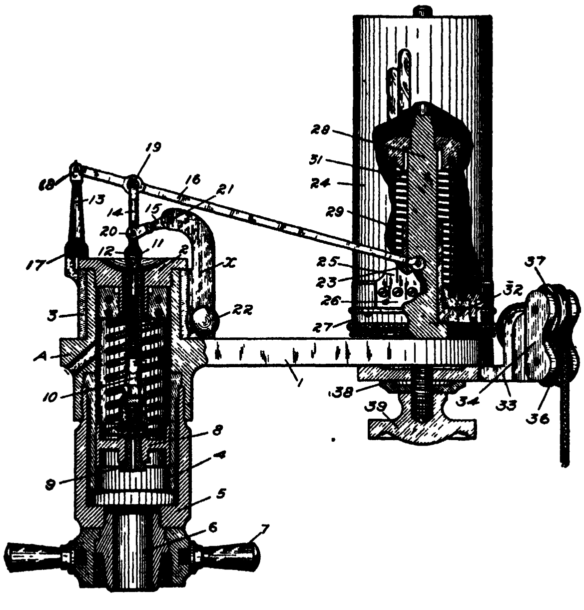


Fig. 2. Part Section of Crosby Indicator

of different diameter on the lower side. The projection with the larger diameter is threaded so that the cap can be screwed into the cylinder. The smaller projection is also threaded to engage with like threads on the spring head which holds it firmly in position. Cap *2* holds sleeve *3* in position in a recess formed for the purpose. This sleeve carries the pencil mechanism, parts of which are *15*, *13*, etc. The arm *X*, which carries lever *15* of the pencil mechanism, is made integral with the sleeve. A handle *22* is provided by which

the pencil point is brought in contact with the paper. This handle is threaded and, being in contact with a stop screw on plate *1*, permits a very delicate adjustment of the pencil point to the surface of the paper on the drum. It is desired to have the pressure just great enough (but no greater) to secure a visible diagram on the paper.

The pencil mechanism, consisting of links *13*, *14*, *15*, and *16*, is a very important part of the indicator. Its essential kinematic principle is that of a pantograph. This mechanism must be so designed and adjusted that the path of pencil *23* is at all times parallel to the path of piston *8*. The links are so proportioned that the movement of the pencil is six times that of the piston.

The indicator card is held on paper drum *24*—which is made of very light metal—by means of clips *25* and *26*. Drum *24* fits on a base *27*, which carries a spring *31* on a central projection *28*; this spring brings the drum back to its initial position when the indicator is detached from the moving parts of the engine or when a return stroke is made.

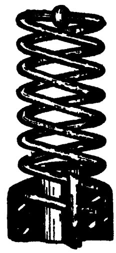


Fig 3
Indicator Piston Spring

The direction in which the cord may be conducted from the drum can be adjusted by means of guide pulleys *36* and *37*, which are attached to the indicator by nut *39* and frame *33*.

The piston spring, Fig. 3, which should be of a good quality of spring steel, must be carefully made and tested, and also carefully handled, as the accuracy of the results depends in a large measure upon the accuracy of the spring.

It will be noted in Fig. 2 that the piston or pressure spring is placed within cylinder *4*, when the indicator is put together for use. This brings the spring in contact with the live steam and, as a consequence, errors may be recorded due to the uneven heating of the spring and contained parts. To eliminate the possible inaccuracies due to heat, the manufacturers have constructed indicators with the spring on the outside, as illustrated in Fig. 4. Aside from the elimination of errors by the use of the outside spring, convenience is obtained in that the spring is more accessible and may be removed or changed without taking the indicator apart, which can not be done with the spring inside. Furthermore, the spring does not get

very much warmer than the surrounding atmosphere, so it is not necessary to allow the indicator to cool before removing it.

Fig. 4 also shows how an indicator may be easily changed from an ordinary steam indicator to an indicator suitable for gas engine work. The change is made by reducing the size of the cylinder to $\frac{1}{4}$ square inch in area and increasing the strength and weight of the pencil mechanism. By these changes, an indicator may be used on gas engines with good results.

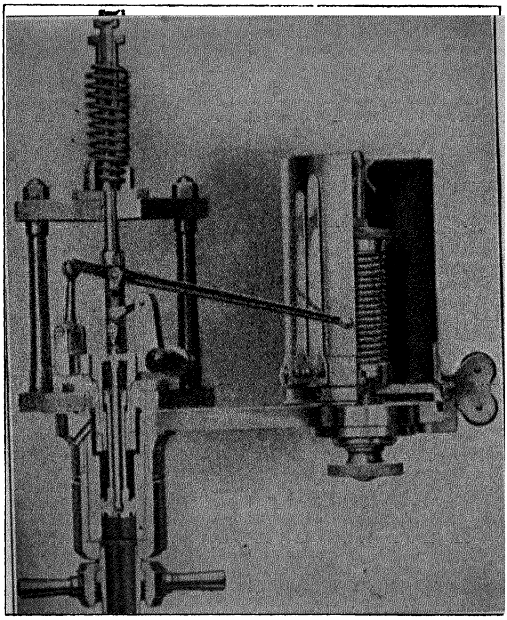


Fig 4 Crosby Indicator with Outside Spring

The Crosby indicator is ordinarily made with a drum $1\frac{1}{2}$ inches in diameter, this size being suitable for high-speed work. If, however, a larger diagram is desired and the speed is low, a drum 2 inches in diameter can be furnished.

Tabor Indicator. The Tabor indicator, Fig. 5, with outside spring, reducing motion, and electrical attachment, is similar in construction, operation, and essential characteristics to the Crosby and other indicators, though there are details of design for which the respective makers claim advantage over other makes. One feature

of the Tabor which is essentially different from the Crosby and the Thompson is its pencil mechanism. As was shown in Fig. 2, the Crosby pencil mechanism consists of a system of levers, which gives

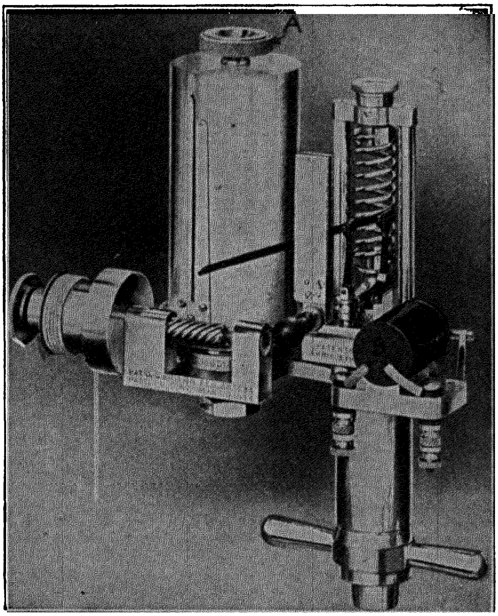


Fig 5 Tabor Indicator with Outside Spring

to the pencil a straight-line motion parallel with that of the piston, with possible slight errors, especially on high cards. As shown in Fig. 6, the scheme for obtaining the parallel or straight-line movement of the Tabor indicator is different from that of the Crosby. It has the connecting links corresponding to *13*, *14*, and *16* in the Crosby, Fig. 2, but in place of link *15*, there is substituted another arrangement. A stationary plate *1*, with a curved slot *2*, is fastened in an upright position to the cap. On the pencil bar is a roller bearing *3*, which is secured to the bar by a pin. This roller moves freely in the curved slot in the guide and controls the motion of the pencil bar. The position of the slot and the guide upright is so adjusted

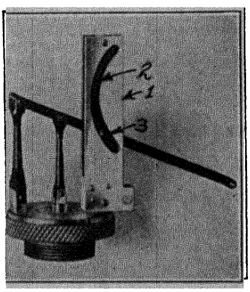


Fig 6 Tabor Device for Straight-Line Movement

and the guide roller is so placed on the pencil bar that the curve of the guide slot controls the pencil motion and absolutely compensates for the tendency of the pencil to move in a curve. There is a minimum of friction in this movement and guide, and no disturbance from inertia has been detected by the most careful tests.

American Thompson Indicator. The American Thompson indicator, Fig. 7, does not differ greatly in general appearance from

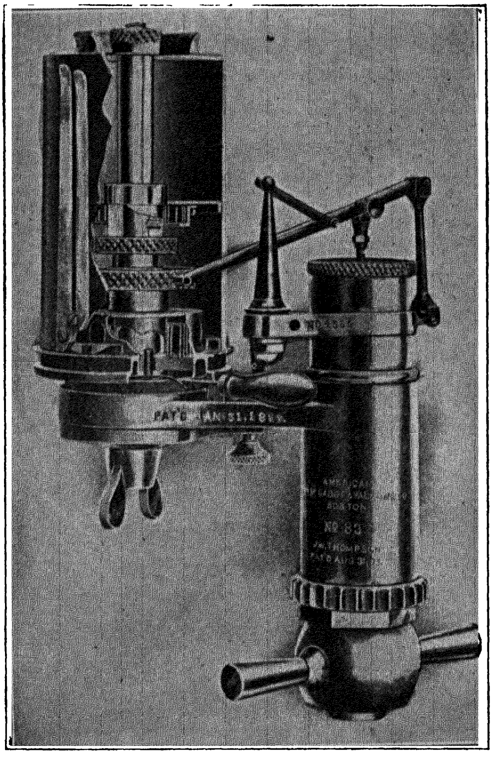


Fig. 7 American Thompson Indicator

other indicators, but a close comparison will show some difference in the details of construction. For instance, it is evident that the arrangement of the levers, which make up the pencil motion, differ slightly from that of the Crosby indicator. Each maker, of course, makes the assumption that his particular arrangement is the best. In most cases, the purchaser must use his own judgment in the matter. Again the connection of the spring to the piston is different on the

TABLE I

Constants of Indicator Springs

Scale of Springs	Maximum Safe Pressures to Which Springs Can Be Subjected — Pounds Pressure per Square Inch with ½ Square Inch Area Piston	
	To 200 Revolutions per Minute	To 300 Revolutions per Minute
8	10	6
10	15	10
12	20	15
16	28	22
20	40	32
24	48	40
30	70	58
32	75	62
40	95	80
48	112	95
50	120	100
60	140	115
64	152	125
80	180	145
100	200	160
120	240	195
150	290	250
200	375	330

Thompson than on others, in that the spring screws directly into an enlargement on the upper side of the piston, thus being rigidly attached to the piston as well as to the pencil motion at the top instead of having a semi-flexible connection by means of a ball and socket joint, as in the Crosby. These two points are the distinguishing ones of the Thompson indicator. The construction of its cylinders, piston, paper drum, etc., are about the same as for those previously described. In the figure, a portion of the drum is shown cut away, disclosing the detent motion, the operation and purpose of which will be described later.

The piston of an indicator is usually .798 inch in diameter, which is equivalent to ½ square inch area. This size piston with springs is designed to indicate pressures up to 250 pounds. When higher

pressures than 250 pounds are used, a piston .564 inch in diameter, representing an area of ¼ square inch, is substituted for the ½-inch piston. This doubles the capacity of the spring and makes it possible to indicate up to 500 pounds.

Since it is the capacity of the spring that limits the height of the indicator card and since the various springs are made to resist a definite amount of pressure, it is necessary that the proper capacity of spring be used at all times. This capacity is designated by the term "scale of spring" which means the amount of pressure required on the piston per square inch of area to raise the pencil point 1 inch. For example, if one hundred pounds steam pressure is being used and a spring having a scale of 40 is placed on the indicator, the height of the resulting card will be $100 \div 40 = 2\frac{1}{2}$ inches. The capacity of the spring is always marked upon it, as 40, 60, etc. The manufacturers of the Tabor indicator recommend the use of springs having capacities for various conditions of speed and pressure as given in Table I.

If an engine is running at a speed not exceeding two hundred revolutions per minute and the steam pressure being used is 180 pounds, the scale of spring to be used is 80. If the revolutions per minute (r. p. m.) be increased to between two hundred and three hundred, then a spring of 80 pounds should not be used for pressures higher than 145 pounds. This table is about the same as that given by other makers of indicators. A common rule for determining the capacity of the spring to be used is to multiply the scale of the spring by 2½ and subtract 15, the result being the limit of pressure to which the spring should be subjected. To illustrate: Assume a spring having a scale of 60. Then $60 \times 2\frac{1}{2} = 150$. $150 - 15 = 135$. Therefore, 135 pounds pressure is the ultimate capacity of a 60-pound spring, which approximately checks Table I.

From the foregoing discussion of the indicator and the study of its construction, it is evident that the essentials of a good indicator are summed up by Professor Thurston in the following paragraphs:

(1) Such form and construction as will insure its meeting the prescribed general conditions—accuracy of representation of variations of steam pressure and simultaneous movement of the piston at all times.

(2) Such simplicity of form as will make it free from liability to accident and failure in operation.

(3) Such lightness of parts and rigidity of whole, as will prevent any inaccuracies of indications arising from its inertia

(4) It should be easily, conveniently, and safely attachable and unmovable and handily manipulated

(5) Stiffness, lightness, and exactness of standardization are prime essentials. Springs should be exactly standard. Moving parts as light as consistent with proper strength and stiffness; stationary parts should be carefully proportioned and rigid.

INDICATOR SPRING TESTING

Apparatus. As the accuracy of the action of the indicator spring is of primary importance in obtaining correct indications,

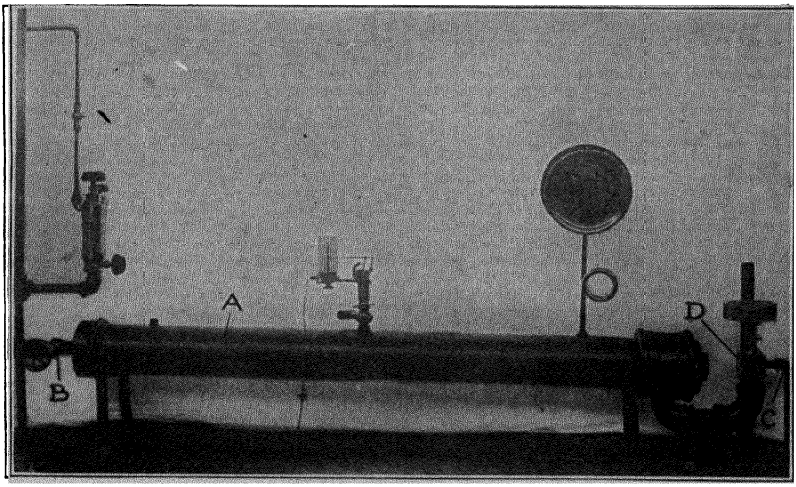


Fig 8 Spring Testing Device

some means must be employed for testing indicator springs. A very simple but efficient apparatus for testing indicator springs is shown in Fig. 8, which consists of a drum *A*, made of 4- or 5-inch extra heavy pipe having steam-tight joints. Steam is admitted to the drum at *B*, and permitted to pass out at *C*, through a piston regulating valve *D*, carrying a disk and weights.

The indicator and a standard test gauge for checking are attached in the manner shown. The pressure regulating valve *D* is very sensitive and responds to a very slight change of pressure. By placing the desired weight on the disk and adjusting valve *B*, the pressure of steam in the cylinder is maintained at a constant value. If

the pressure should rise higher than desired, the piston valve D rises, permitting the escape of steam through pipe C, and in this way maintains a constant pressure in the drum.

Spring Calibration. To test or calibrate the spring, proceed as follows: Put the indicator together properly and see that the

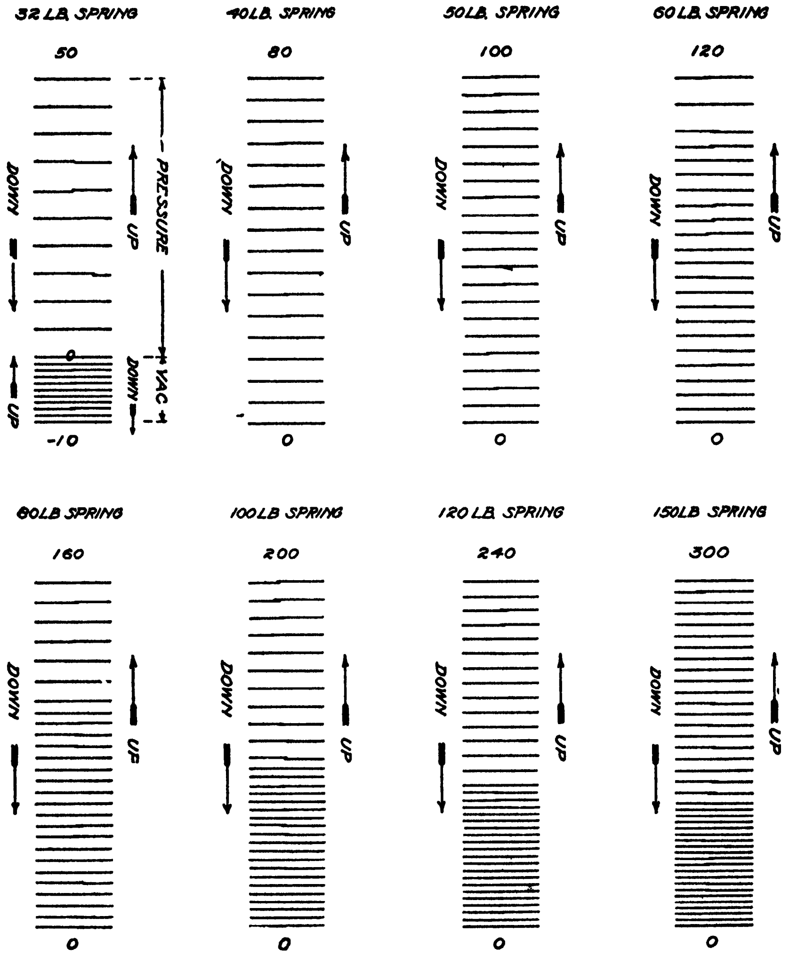


Fig 9. Cards Showing Records of Spring Tests

piston is oiled and in place. Attach the indicator in the usual manner. After the indicator has been warmed up by permitting steam to act on it, put the desired weight on the disk and spin it, at the same time moving the indicator drum by pulling the cord and hold-

ing the pencil against the paper drum, thus recording the pressure on the paper. Proceed in this manner by equal increments of pressure until the capacity of the spring has been reached, then reduce the pressure by the same increments until zero is reached. The operation of taking both the upward and the downward readings should be continuous, stopping only long enough to change the weights and to make the proper indications.

Measuring the Cards. After the cards have been taken, they may be measured by means of a scale in the usual manner. The pressures thus measured should check within a fairly close margin of the readings corresponding to the gauge and the tester.

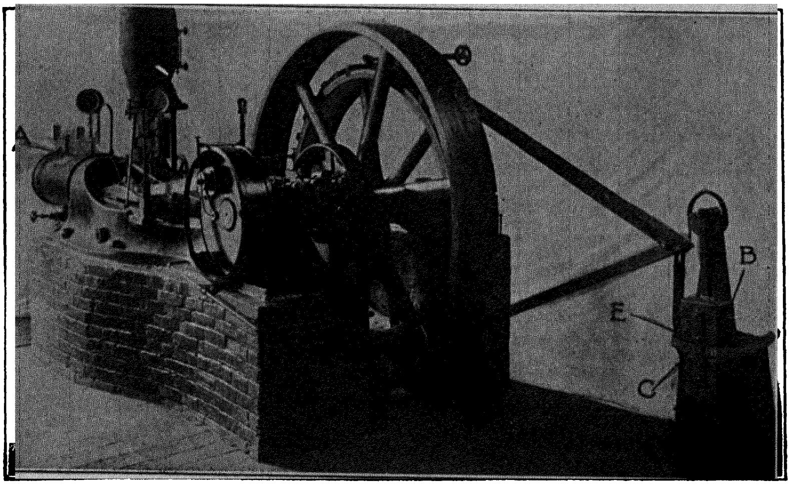


Fig 10 Engine Showing Two Indicators Screwed Into Cylinder

The cards, Fig. 9, show the records obtained from tests of various springs. It is evident that some of the records taken with increasing pressures do not coincide with the corresponding record when going down or with decreasing pressures. For very accurate work, the spring should be used with the piston and in the indicator with which it was tested.

Engine Connection. The attachment of the indicator to the engine should be such that the pressure of the steam on the indicator piston is exactly the same as that acting at the same instant on the engine piston. In order to secure this result, the steam connection between the indicator and the engine should be amply large and direct.

If possible, the indicator connection should be screwed directly into the cylinder as shown at *A*, Fig. 10. In making this attachment, a hole is drilled in the cylinder and a connection is made to the indicator by means of a standard $\frac{1}{2}$-inch pipe and a proper valve or cock. The hole in the cylinder should be drilled in the clearance space, where the piston will at no time cover any portion of the opening, and where no strong currents of steam will sweep directly into the passage.

If it is possible to remove the cylinder heads, it should be done before drilling, so as to properly locate the holes and to remove any chips which may happen to fall into the cylinder while it is being

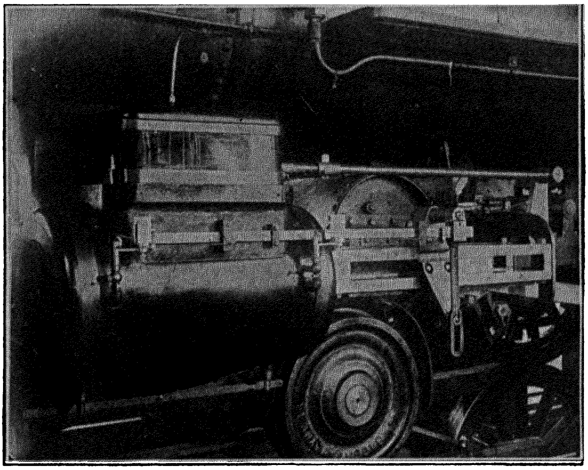


Fig 11 Method of Attaching Indicator to Locomotive

drilled. If it is not feasible to remove the cylinder heads, the cylinders should be carefully blown out with steam before running the engine, as much damage may result from the chips in the cylinder The indicators should be attached in an upright position if possible. It is best to have an indicator attached to each end of the cylinder, so that cards may be taken simultaneously from both ends. Before drilling the holes, a general plan or scheme should be studied out for the attachment of the indicator and its necessary appliances, as the type of engine (whether vertical or horizontal), the type of crosshead, and the necessary room for operation may be quite different for each case; so a strict rule can not be laid down for this procedure.

Suffice it to say that generally the indicator can be attached to the side of the cylinder or to the top, as shown in Fig. 10. Figs. 10 and 11 illustrate the methods used in attaching indicators to a simple engine and a locomotive, respectively. It will be observed that all of these connections are short and direct, that the indicators are in an upright position, and that the cord of the indicator is led straight to the crosshead connection.

Sometimes it is not convenient to use two indicators or it may be that the engineer does not care to bear the cost of two, so only one is used. When only one is used, a pipe leading from each end of the cylinder is connected to the indicator by means of a three-way cock, as

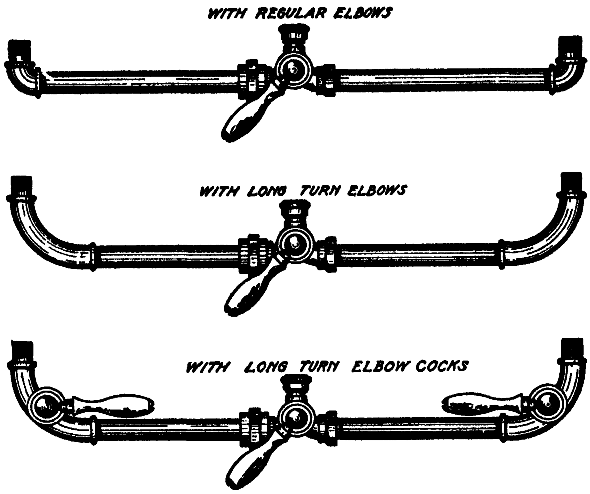


Fig 12 Typical Three-Way Valve

shown in Fig. 12. By the use of this cock, the indicator is put in connection first with the head end (h. e.) and then the crank end (c. e.) of the engine. This should be done with as little loss of time as possible so that the cards will represent, as nearly as possible, actual conditions of pressure in the cylinder. The three-way cock, or any other cock which may be used in the system, should have an opening as large as that in the cylinder connection of the indicator. It should also have a hole in one side for the purpose of freeing the indicator and its connections from any water that may come over with the steam.

Avoid Long Pipe Connection. Long connections between the indicator and the engine cylinder should be avoided in all cases.

Experiments conducted at Purdue University have demonstrated the fact that any pipe connection between the indicator and the engine is likely to affect the action of the indicator. Under ordinary pressures and speeds, a length of pipe over 3 feet in length so distorts the card that the results obtained are useless except for approximate work.

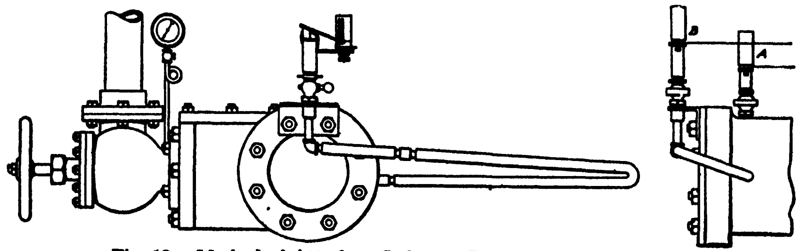


Fig. 13 Method of Attaching Indicator Piping to Engine Cylinder

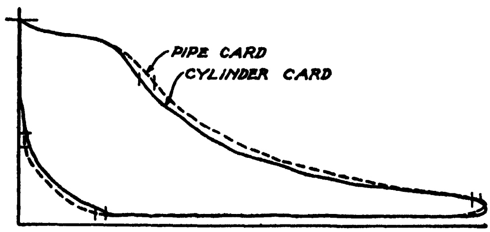


Fig 14 Indicator Card Showing Effect of Changing Connecting Pipe

In order to determine the effect of long pipe connections between the indicator and engine, upon the form of the cards, a series of tests were conducted in the Engineering Laboratory of Purdue University under the direction of Dean W. F. M. Goss. The results of these experiments formed the basis of a paper which Dean Goss

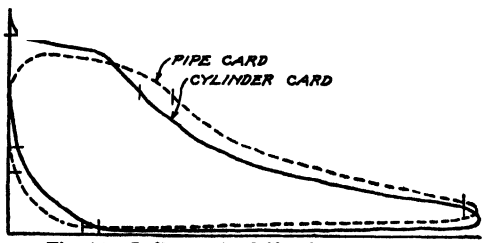


Fig 15. Indicator Card Showing Effect of Changing Connecting Pipe

presented before a meeting of the A. S. M. E. at St. Louis in May, 1896. The experiments were made in connection with a $7\frac{3}{4}$- by 15-inch Buckeye engine. Very great care was taken in the selec-

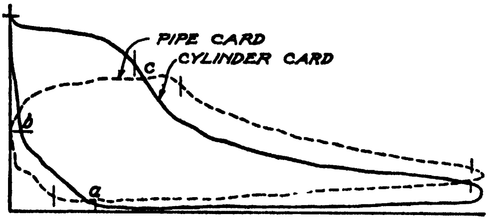


Fig 16 Indicator Card Showing Effect of Changing Connecting Pipe

tion and testing of the indicators and in their manipulation, in order to insure that any distortion which might occur in the cards would be due entirely to the pipe connections. The indicators were

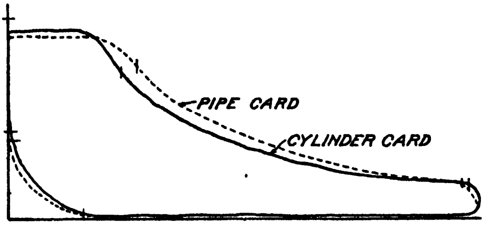


Fig 17 Indicator Card Showing Effect of Changing Connecting Pipe

attached to the cylinder, as shown in Fig. 13, both being connected at the same end so that the indicator pistons would be exposed as nearly as possible to identical conditions.

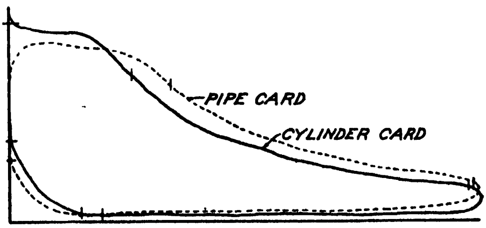


Fig 18 Indicator Card Showing Effect of Changing Connecting Pipe

The indicator *A* and the cards obtained therefrom will be hereinafter designated as cylinder indicator and cylinder cards and it is

assumed that this indicator will give indications true to the conditions in the cylinder.

The indicator *B* and the cards obtained therefrom will be designated as the pipe indicator and pipe cards, respectively, and it is assumed that any perceptible difference in the cards obtained from

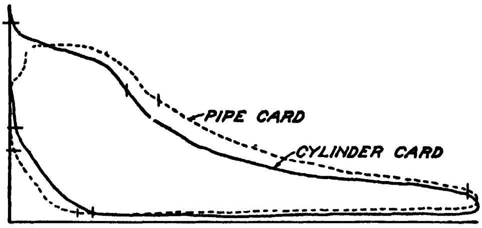


Fig 19 Indicator Card Showing Effect of Changing Connecting Pipe

the cylinder indicator and from the pipe indicator will be due entirely to the pipe connections.

Pipe connections of 5, 10, and 15 feet were used, the length of pipe being measured from outside of the cylinder walls to the end of the coupling under the indicator cock. Care was taken in securing easy bends in the pipe so as not to retard the action of the steam.

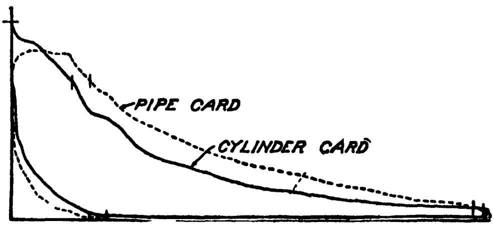


Fig 20 Indicator Card Showing Effect of Changing Connecting Pipe

The pipes were also properly insulated in order to avoid in so far as possible any condensation.

The method of procedure was to run the engine for a short length of time, until the desired speed, cut-off, pressures, etc., were obtained, then cards were taken simultaneously from the two indicators. Two cards were taken from each indicator, then the indicators were interchanged and two more cards taken from each, thus obtaining four cylinder cards and four pipe cards.

Figs. 14 to 22 inclusive illustrate the effect of the pipe on the form of the indicator card with the engine running under various

conditions. A study of these cards reveals the fact that the length of the piping to the indicator affects very materially the area of the diagram. The events of the stroke, while remaining unchanged, are apparently generally made later by the use of the long pipe, but in some cases, some of them are earlier.

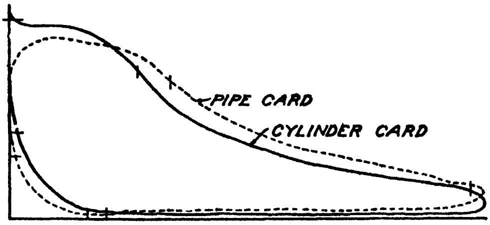


Fig 21. Indicator Card Showing Effect of Changing Connecting Pipe

It is hoped that what has been said is sufficient to point out the importance of the short indicator connection.

Continuous Diagrams. From the study of the indicator, it has been obvious that the indicator card gives an indication of what is taking place in the cylinder at a specific moment. This being the case, it is practically impossible to obtain by its aid determinations

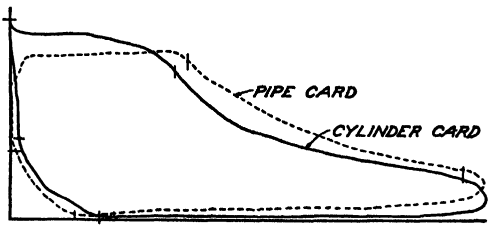


Fig 22 Indicator Card Showing Effect of Changing Connecting Pipe

that are to be relied upon when the engine is working under constantly varying conditions, as in gas engines, locomotives, marine engines, and rolling-mill engines. To meet this demand, an attachment has been developed whereby it is possible to take a continuous card, thus getting exact determinations under the most variable conditions.

Crosby Device. Fig. 23 represents a Crosby indicator equipped for taking continuous diagrams. The special drum is designed so as to be applied to any Crosby indicator, and uses a roll of paper

2 inches wide and 12 feet long upon which the series of diagrams are traced. The roll of paper is located within an opening in the drum. From the roll, the paper passes around the outside of the

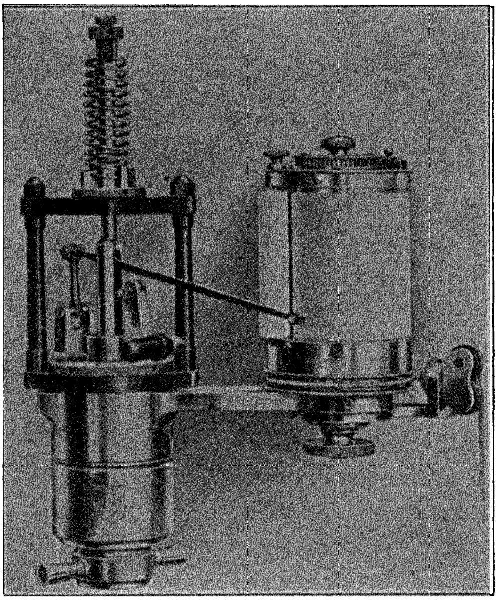


Fig 23 Crosby Indicator with Continuous Diagram Device

drum, thence inward to a central cylinder to which it is attached. In taking cards the paper rolls up on the central cylinder, which is concentric with the drum, and may be withdrawn through the top and easily detached. On the top of the drum is a knurled head,

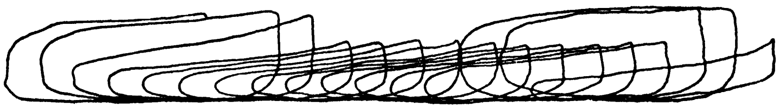


Fig 24 Continuous Diagram from Rolling-Mill Engine

loosely attached to the drum spindle, which controls the distance between diagrams. Adjustment can be made so that from 6 to 100 diagrams can be made per foot of paper.

Fig. 24 illustrates a series of continuous cards taken from a rolling-mill engine and clearly shows the widely varying conditions.

Fig. 25 shows cards taken from an automatic cut-off rolling-mill engine.

After providing proper means for attaching the indicator to the cylinder, the next important step is to provide a convenient and at the same time correct reducing or drum motion.

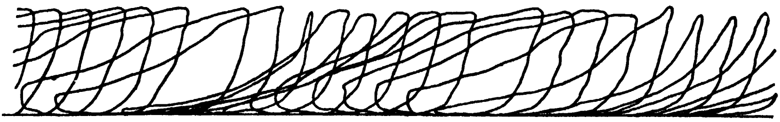


Fig. 25. Continuous Diagram from Automatic Cut-Off Rolling-Mill Engine

Reducing Motions. In the description of the indicator, it was noted that the indicator card is held on the circumference of the paper drum by means of clips. Since the circumference of the drum is much less than the length of stroke of the engine, some means must be provided to reduce the motion of the drum. As each engine and its location will be different, no strict rule can be given whereby one can at once provide a reducing motion, but each case must be studied and the best means possible provided to meet the exigency. A few examples and principles will be given and doubtless they will suggest others to meet specific cases.

Brumbo Pulley. The Brumbo pulley, Fig. 26, is easily and quickly made and can be attached to almost any engine. If it is to be used for only a short time, it may be constructed of wood, care being exercised in having close fitting joints. If the engine is to be indicated frequently, it is better to make the reducing motion of metal in order that the wear in the joints may be minimized. The reducing lever A is pivoted overhead to some temporary support or, if it is to be permanently attached to the engine, some permanent support, such as an upright post or bracket, may be attached to the frame of the engine, as in Fig. 10. The segment S is made fast to the lever A, so that its semicircumference is true with its pivot

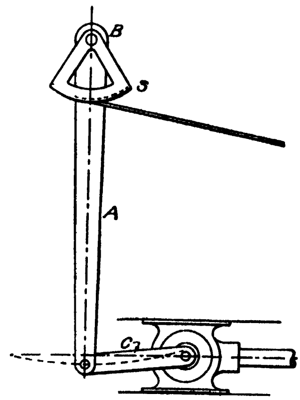


Fig 26 Brumbo Pulley Reducing Motion

point *B*, upon which the lever swings. The sector may be set at any angle with the lever. The lower end of the lever *A* is attached to the crosshead through the link *C*. The length of the lever *A* should be at least one and one-half times the length of stroke of the engine. The length of the connection *C* may be about one-half of the length of stroke, but it may be greater. When the crosshead is at its mid-position, the lever *A* should be vertical. During the stroke of the engine, the link *C* should swing equally above and below a horizontal position.

With this form of reducing motion, the cords may be led in any direction in a vertical plane from the sector and one or more cords may be led off to different indicators. The face of the sector should be true and the radius must have a value sufficient to give the required motion to the drum. To fulfill this condition, the radius of the sector must bear the same relation to the length of the lever *A* as the proposed length of the indicator diagram bears to the stroke of the engine.

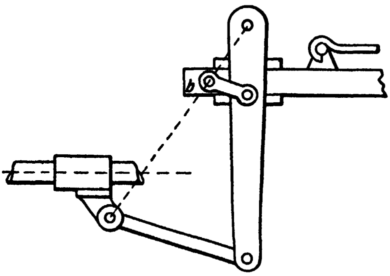


Fig 27 Diagram Showing Principle of Brumbo Pulley

EXAMPLE Suppose it is desired to make a reducing motion for a 10- by 16-inch engine. Assume the length of the lever *L* to be 24 inches and the required length of card *D* to be 4 inches. The length of stroke *S* is 16 inches. Designating the radius of the sector as *R*, then

$$R : L :: D : S$$

Solving for *R* in the above engine

$$R : 24 :: 4 : 16$$
$$16\ R = 96$$
$$R = 6 \text{ inches}$$

The principal objection to the Brumbo pulley is that it is not interchangeable, that a different one is required for engines of different types and sizes. If it is carefully made and attached, however, it will give results with very slight inaccuracies. The design illustrated in Fig. 27 is theoretically correct, and its construction and operation are so clearly shown that a detailed description is not deemed necessary. It has been successfully used for experimental work in colleges and universities for a number of years.

Pantograph. Another form of reducing motion, known as the pantograph, is shown in Fig. 28. It is placed horizontally, with the pivot *B* resting on a support opposite the crosshead when in mid-position. The pivot *A* is attached to the crosshead, usually by having the stud *A* inserted in a hole drilled in the crosshead. If the pivot *B* is adjusted to the proper height and at the right distance from the crosshead, the cord from the indicator may be attached to the pin *E* without any pulleys, which is very desirable. The length of the diagram is adjusted as desired by means of the movable piece *C D* and the pin *E*. The pin *E* must always be on a line joining the pivot points *A* and *B*. The pantograph gives correct results when in good condition and properly attached but, on account of the large number of joints, it may become unsatisfactory. This type is usually used on engines having a long stroke and where it is not convenient to attach a Brumbo pulley or its equivalent. It is applicable to all types of engines of any length of stroke. In two of the three forms of reducing motions just described, there are chances for inaccuracies. There is an error in the Brumbo pulley which may or may not be very large, depending on the proportions of the levers, and there may be lost motion in the many joints of the pantograph. The inaccuracy of the Brumbo pulley can be materially decreased by using, instead of the sector, a *sliding bar*, as in Fig. 11. This sliding bar is supported by means of brackets and is attached to the lever by means of a pin which works in a slot similar to that at the lower end of the lever. As the crosshead moves to and fro, the sliding bar likewise moves, and its motion is proportional to that of the crosshead at all points of the stroke. All things being considered, the principle of the reducing motion shown in Fig. 11 is all that could be desired; it is especially suited for locomotive engines.

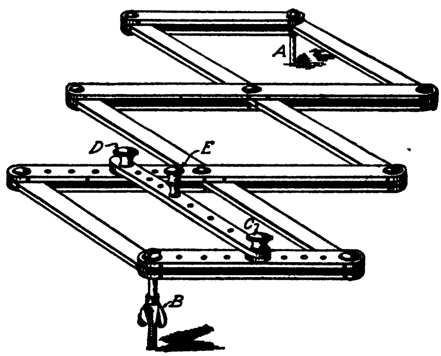


Fig 28 Pantograph Device for Reducing Motion

Reducing Wheel. Nearly all makers of indicators manufacture

a reducing wheel apparatus which serves the same purpose as the lever and pantograph types of reducing motions. A type of this apparatus is shown in Fig. 29; also it may be seen in Fig. 5 attached to the indicator. The following description is given by the makers.

The reducing wheel is composed of a supporting base piece *A*, provided with short standards *B* that form bearings for the worm shaft on which the flanged pulley *D* is rotated, the outer bearing being a pivot which receives the entire thrust of the shaft, thus reducing the friction to a minimum. It is connected directly to the indicator upon the projecting arm that supports the paper drum, and the teeth of the worm shaft mesh directly into the teeth on the drum carriage *C*. Connected with the base piece is a spring case *E*, and on the extreme end of the worm shaft is secured a collar *F* through which freely slides a clutch pin, one end of which is securely fastened to a thumb piece for operating it.

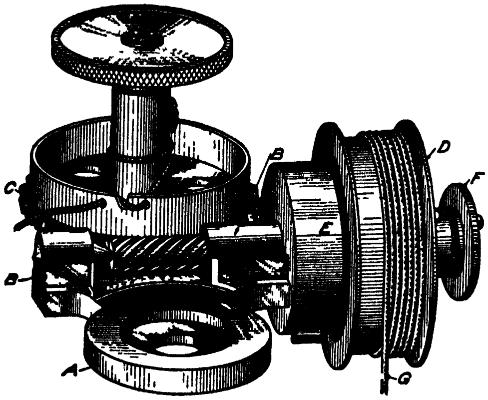


Fig 29. Tabor Reducing Wheel

The flanged pulley *D* runs freely and independently on the worm shaft, and has on its outside a clutch-shaped hub. To this pulley is connected the actuating cord *G*, which should encircle it a sufficient number of times to have its length, when unwound, a little more than equal the length of the stroke of the engine. The other end of the cord is secured to the crosshead of the engine or to a standard bolted thereto or to any moving part that has an exact similar motion, and must be connected in line from the pulley.

Enclosed in the spring case *E* is a small, plain, spiral steel spring which operates solely to return the pulley to its starting point, after it has been revolved in one direction by the forward movement of the crosshead. As this pulley has an independent rotating back-and-forth motion on the worm shaft, the necessity of unhooking the cord when the indicator is not being operated is entirely overcome.

The paper drum is rotated forward by means of the pulley through its worm shaft, engaging with the worm gear on the paper drum carriage, and is rotated in the opposite direction by the action of its own retracting spring. On top of the paper drum is a knurled thumb piece (see *A*, Fig. 5) made with a projecting pin on its under side to engage with a similar pin located in the top of the drum; this is to be used by the operator in moving the paper drum slightly forward, preparatory to taking a diagram, in order to prevent it from striking against its stop on the return motion.

To operate this device, first, select a pulley whose diameter is about one-twelfth of the length of the engine stroke in inches. Properly place this pulley upon the worm shaft by removing the clutch and then sliding the pulley onto the shaft, being particular that the small hole in the pulley brass disks sets over the projecting pin in the cover of the spring case. Then replace the clutch by pushing it onto the shaft as far as it will go, and secure it there by means of the set screw.

Now place the indicator on the engine in such a position that the side of the pulley *D* will be parallel with the motion of the crosshead. Run out the loose end of the cord to a distance of at least 12 or 18 inches beyond the extreme forward travel of the crosshead, still leaving a turn or two of the cord on the pulley unwound. While holding the cord, allow it to gradually recede and rewind itself on the pulley until its loose end has reached a point coincident with the extreme backward travel of the crosshead. If only a slight tension of the cord exists at this point, it will be sufficient, and the cord may then be attached to the selected point on the crosshead. The cord tension may always be adjusted either by winding the cord on, or unwinding it from, the pulley, as the case requires, one increasing and the other decreasing the tension.

A much lighter cord can be used in proportion as the sizes of the pulleys increase.

When the crosshead, with cord connected, is at its extreme forward travel, there should be just sufficient tension on the spring enclosed in the spring case to take up all slack of the cord when running, without overtaxing the spring. If, upon starting the engine, the cord should at first run unevenly on the pulley, turn the indicator to one side slightly until a perfect and uniform winding of

the cord is obtained, which can always be easily secured. When the pulley is running, motion to the paper drum is obtained by pushing in the swivel collar to which the clutch pin is secured.

When ready to take diagrams, after placing the paper on the drum it is necessary first to advance the drum away from its stop fully ¼ inch, which can be done by turning with one hand the knurled top thumb piece. While holding drum in this position, with the other hand push in gently the swivel collar to start the paper drum in motion. The motion of the paper drum can at any time be stopped for removing diagrams taken and renewing the paper by withdrawing the swivel collar or by turning the top thumb piece, the latter method being preferable, as it prevents damage from too severe contact with the paper drum stop. The stopping of the paper drum will not affect the motion of the pulley, which will continue to revolve independently while the engine is in motion until the cord is disconnected.

With the indicator are usually furnished three different size pulleys. Unless otherwise specified, pulleys furnished are 1, 2, and 3½ inches in diameter. These pulleys are sufficient for the average work required of an indicator Larger sizes can be obtained if needed.

The reducing wheel form of reducing motion has many points of advantage over the pantograph and lever in that it is conveniently attached and it allows the operator to start and stop the motion of the paper drum without disconnecting the cord where attached to the crosshead, which is an annoying thing to do under some circumstances. The reducing wheel does not work under high speeds as satisfactorily as for the lower speeds, which is, perhaps, one of its most objectionable features. When the indicator is kept in constant use for several hours at a time, some trouble may be experienced with the cord becoming worn and breaking during a test.

From this study of the reducing motion, it is evident that much must be left to the discretion of the operator as to the selection and attachment of the motion.

Crosby Reducing Wheel. The Crosby reducing wheel, shown in Fig. 30, is attached directly to the cylinder cock of the steam engine, and has connected to it the steam engine indicator which it is to serve; thus it forms a base of support for the latter, and receives all the strains and shocks in the operation of the engine, to

the relief of the indicator. Its bearings are made comparatively frictionless by the introduction of minute balls running in hardened tool steel races, thus affording lightness and freedom of movement. The cord pulley is horizontal, to allow the cord leading to the engine crosshead to take any direction the circumstances may require, without regard to the position of the indicator.

Whenever the reducing wheel is to be attached to a vertical engine, an elbow nipple is provided, which will allow the cord pulley

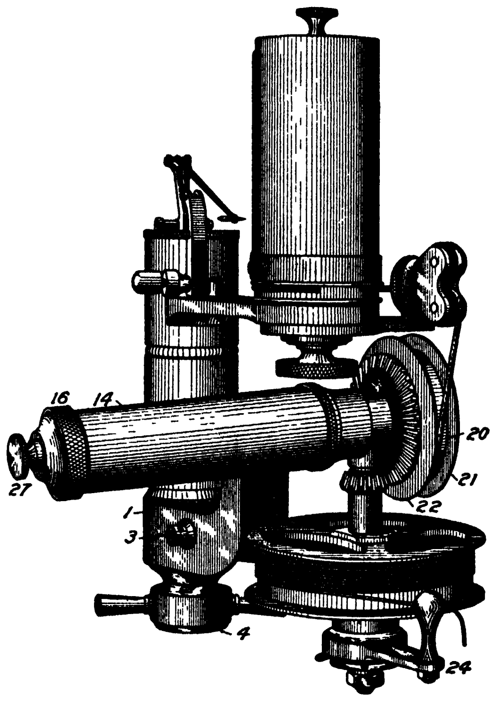


Fig. 30. Crosby Reducing Wheel Attached to Indicator

to travel in the proper plane for guiding it to the crosshead of the engine with the indicator in an upright position as usual.

OPERATION RULES

The Crosby reducing wheel is attached directly to the cylinder cock of the steam engine by means of union *4* of standard *1*. Connect the indicator to standard *1* with the paper drum standing over spring *14* and the indicator guide pulley in a proper position over stroke pulley *20*.

To attach the cord guide: Loosen cord guide *24* by means of the screw beneath the cord pulley; then move it around to the proper position for the cord to pass directly through the hole in the cord guide without rubbing, to the crosshead of the engine and tighten it in place.

To take up the tension spring: Release thumb screw *27* in the end of the shaft within spring tube *14*; withdraw knurled spring head *16* from its square end, and turn it one or more squares as may be desired.

To adjust the stroke pulley: Remove knurled disk *21*, which holds in place stroke pulley *20* on the gear shaft; place on the shaft the stroke pulley desired; replace the disk and screw it up firmly.

To attach the indicator cord: Wrap the indicator cord one or more turns around stroke pulley *20*, passing the end through the hole in, and around, the hook of knurled disk *21*.

When used with other indicators, loosen bolt *3* in the side of standard *1*, where it is attached to the cylinder cock of the steam engine; remove the bushing and insert another fitted to the indicator to be used.

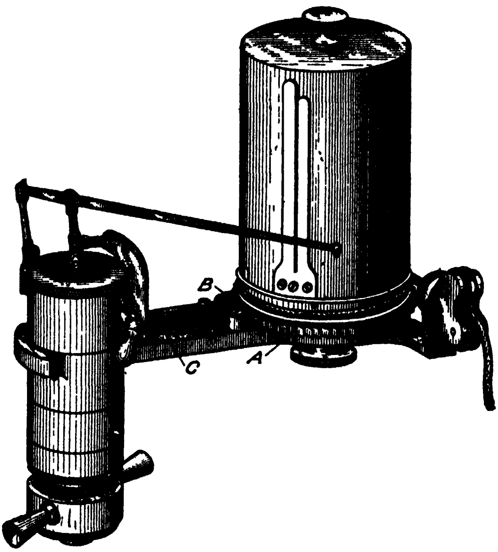


Fig 31 Indicator with Detent Attachment

Simultaneous Indicator Cards. In making complete and reliable tests of power plants, it is desirable that all of the cylinders of the compound and multiple engines be taken simultaneously at a given signal. This requirement, if the indicators are hand-operated, would necessitate an operator at each cylinder, which would be expensive and besides would not insure the simultaneous taking of the cards. The makers of indicators have met this need by supplying the market with an electrical attachment *B*, Fig. 5, which is attached to each indicator that is to be used. It is not thought necessary to give a description of the construction and operation of the appliance, but suffice it to say that by pressing an electric button, the pencils of all the indicators in the circuit are simultaneously brought in contact with the paper and thus a record is made.

Detent Attachment. Another attachment that can be obtained and which is of much convenience to the operator at times is the detent attachment, shown in Fig. 31. It consists of a ratchet *B* that fits into the teeth of the wheel *A*. When the operator wishes to stop the motion of the paper drum, he pushes the lever *C*, which causes *B* to come in contact with *A*, as shown. Thus, the operator may change the card, and do other things without disconnecting the indicator cord from its crosshead connection. This is a very desirable attachment when indicating high-speed engines and when taking cards on a locomotive on the road where conditions are not ideal for using the indicator.

Fig. 32, illustrates the detent attachment used on the Thompson improved indicator. With this new improved detent motion, in order to stop the paper drum it is only necessary to move lever *A* in the direction traveled by the paper drum until the drum releases itself. The lever must then be returned to its original position. When ready to take the diagram, turn forward the paper drum, by means of the milled rim *B* on top, until it catches, causing the drum to revolve in the usual manner; then take the diagram and release the drum as described above. Before taking the diagram, see that the parts are cleaned and well oiled. To oil, remove the knurled nut *F*, take off the paper drum, then with the wire clip (which is sent with each indicator) remove the auxiliary spring case *H* by catching the end of the clip in the notches of the spring case, turning it forward until it releases from the catches; then move the spring and inner sleeve *I*. After cleaning and oiling, replace the inner sleeve *I* by inserting it into the drum so that the pin on the outside of the sleeve will enter the slot inside of drum bearing, and turn it until it comes to a stop; then with the wire clip catch hold of the auxiliary spring holder *H* and give the auxiliary spring *E* a tension of about one-fourth turn, and catch the points on the spring case *H* into the slots provided for them.

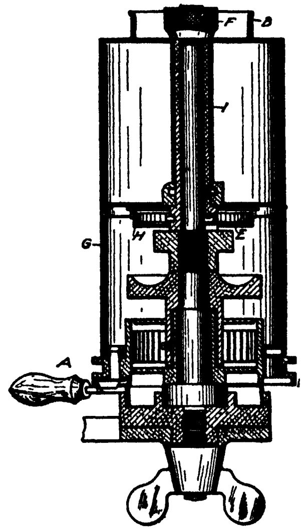


Fig 32 Thompson Detent Attachment

ASSEMBLING AND ADJUSTING THE INDICATOR

Thus far, it has been the endeavor to mention the chief features of different makes of indicators and to point out the important points to be observed in the attachment of the indicator for obtaining a correct movement of the paper drum. Before proper results can be obtained, however, it is absolutely necessary that the indicator be properly put together. By way of illustration, reference will be made to Fig. 2, which shows the Crosby indicator with all of the parts connected together ready for connection to the engine cylinder.

Assembling Crosby Indicator. When the indicator is removed from the engine cylinder, the spring should in every case be disconnected from the piston and well cleaned before putting away. To remove the piston and spring, unscrew cap *2;* then take hold of sleeve *3* and lift all the connected parts from the cylinder. This gives access to all the parts for the oiling and cleaning, which should be thoroughly done after each time the indicator is used. None of the pins as *17*, *19*, etc., should be removed except in making repairs and they should be kept well oiled in order to reduce friction. After removing the spring and cleaning the indicator thoroughly, connect the parts together, leaving out the spring, and put the indicator away. It is evident that the indicator must be put together each time it is used and in doing this great care must be exercised in order to insure satisfactory working of the parts. It is important to notice that on the under side of the shoulder of the piston rod *B*, Fig. 33, there is a circular channel formed to receive the upper edge of the slotted socket of the piston *A*.

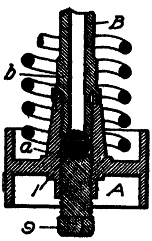


Fig 33 Placing Indicator Spring

Connecting Piston Rod. Whenever it is desirable to connect the piston rod with the piston, either in the process of attaching a spring, or for the purpose of testing the freedom of movement of the piston in the cylinder without a spring, *be sure to screw the piston rod into the socket as far as it will go;* that is, until the upper end of the socket *a* is brought firmly against the bottom of the channel *b* in the piston rod. This insures a perfectly central alignment of the parts and, therefore, a perfect freedom of movement of the piston within the cylinder.

Attaching Spring. To attach the spring, place the piston rod *B*, Fig. 33, in a hollow wrench provided for that purpose, so that the wrench will encircle the hexagonal part of the piston rod. Holding the hollow wrench with the piston rod in place, in a vertical position, place the spring over the wrench so that round head *1* will be in the concave portion of the end of the piston rod. Unscrew set screw *9* until it is almost removed from the piston. Invert the piston and insert the transverse wire at the lower end of the spring in the slot in the socket of the piston. Screw the piston on to the piston rod as far as it will go. Remove the wrench, and holding sleeve *3* and cap *2* (see Fig. 2) in an upright position, so that the pencil lever will drop to its lowest position, engage the threads of swivel head *11* with those inside the piston rod, and screw it up until the threads on the lower projection of cap *2* engage those in the spring head. Continue the process until the spring is screwed up tightly on cap *2*. After this, holding sleeve *3* in one hand, with the other turn the piston swivel onto the piston rod. It sometimes occurs that when the piston rod is screwed up on the swivel, the atmosphere line (the line inscribed by the pencil on the paper drum) is not properly located, so it becomes necessary to unscrew the piston rod until the atmosphere line is at the proper height. A little practice will soon teach one about how the parts should be left in order to bring the atmosphere line in the correct position. It is important to avoid having the atmosphere line too high, as trouble might result if the pencil point moved above the top of the paper drum.

Having thus secured the spring to the piston and cap, take the open wrench and turn set screw *9* snugly against the head on the spring. It is important that this should not be done until the spring has been securely fastened to the piston and to cap *2*, for there is then less likelihood of error in alignment. After completing the successive steps named, oil the piston with a good cylinder oil, insert it in the cylinder, screw cap *2* down tightly, which will cause all of the parts to assume their proper places.

Testing Action. Before placing the spring on the indicator, it is well to test the indicator in order to determine whether or not it is in good condition. To do this, put the indicator together carefully, omitting the spring, oil the piston, and place the parts in the cylinder.

Then raise the pencil as high as it will go and release it. If it returns to the bottom of its own accord, it is an assurance that everything is in alignment and that there is little friction in the moving parts.

Adjustment. *Length of Indicator Cord.* Having carefully connected the parts of the indicator and attached it to the engine cylinder in the proper manner, the next step is to adjust the length of the indicator cord, which should be as short as possible. If it is impossible to use a short cord, then a fine steel or brass wire should be used. The builders of indicators usually furnish a braided cord which has been well stretched and which gives good results. Sometimes it is convenient to make a loop in the end of the cord, which is engaged by a small hook attached to the reducing motion. One method for adjusting the length of the cord is as follows: The hook *A*, Fig. 34, is attached to the indicator cord. The cord *B* from the

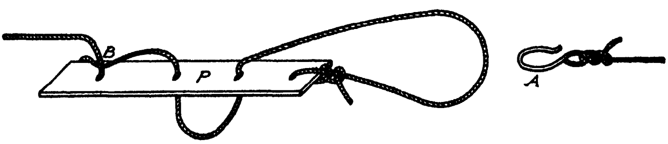


Fig 34 Device for Altering Length of Indicator Cord

reducing motion which passes through the holes in the plate *P*, as shown, is adjusted in length by slacking it at the point *B* and slipping the plate along the cord. To avoid an accident, in the way of injuring the indicator or the reducing motion, it is best to determine as nearly as possible the necessary length of the cord before hooking up to the reducing motion. To determine the length of the cord, take hold of the end of it, and the hook to which it is to be attached; holding them in their relative positions, follow the motion of the reducing lever, keeping the cord tight, thus pulling the drum from one stop to the other, observing if the string must be lengthened or shortened to insure the drum traveling the proper distance. Having determined the length of the cord, hook the two cords together and ascertain whether or not the indicator drum strikes the stop at either end of the stroke.

Adjusting Card and Pencil. Having made the proper adjustment of the length of cord, put the indicator card on the paper drum, being sure that it is smooth and even, as otherwise the diagram will

not be a true representation of the pressure in the cylinder. Considerable practice is required before one can put cards on the drum smoothly and rapidly, but this is desirable. After having caught both ends of the card by the clips, bend the ends over so that they will not interfere with the pencil arm.

Adjust the pencil stop so that the pencil can bear only very lightly on the paper when in the proper position. Always use a pencil with a smooth sharp point, so the lines obtained will be plain and fine. A fine pointed pencil produces less friction when in contact with the paper, which is desirable. The pencil may be easily sharpened by using a small piece of sandpaper or a file.

TAKING CARDS

Everything being in readiness, attach the cord to the reducing motion and, with the pencil off of the card, open the cock and let steam pass into the cylinder of the indicator for a few strokes to warm it up; then put the pencil in contact with the paper for one revolution, after which turn the cock so that no steam is in the cylinder. Again hold the pencil point in contact with the paper, thus getting the atmosphere line. The atmosphere line should always be taken last, in order that there is assurance that all the parts of the indicator are of the same temperature.

Condition of Indicator. It is important to know that the indicator is working properly at the beginning of the test, so after taking the first card, the indicator may be tried to see if it is working correctly. Open the cock and let the piston make a few strokes, close the cock, place the pencil in contact with the paper, at the same time turning the drum by hand; if the pencil retraces the original atmosphere line, or if after a slight pressure up or down on the piston, the pencil returns to the atmosphere line, it is evidence of its being in good condition. If the indicator fails to do the above, the pencil movement is not free in its joints, there is lost motion, or the piston does not move freely in the cylinder of the indicator.

A card should be taken from both ends of the cylinder, for there can be no positive assurance that the same conditions exist in both ends. In fact, oftentimes there is quite a difference between the cards for the head end (h. e.) and the crank end (c. e.), due to inaccuracies of the valve motion and other defects.

Sample Indicator Card. The cards are made of a good grade of white paper, one side being finished smooth. It is on the smooth side that the diagram is made. Manufacturers frequently furnish blank indicator cards having printed on the back of each a set of blank spaces to fill out, which is convenient for filing for the purpose of preserving important data. All of the blank spaces may not be filled out each time, but the more important points should never be neglected. The following form has been used by different successful engineers:

Date . .	Name of operator ..	Builder of engine . . . .
Time	Owner of engine .	Kind of valve motion . . . .
Diam. of cylinder .	Kind of work done by engine..	Kind of steam valves . . . .
Length stroke		Kind of exhaust valves . . . . .
R.P.M	Remarks ..	Kind of condensers . . . . .
Speed of piston		Kind of heater. . . . . .
Diam piston rod		Kind of boiler . . . . .
Area steam port	Barometer reading. .	Kind of fuel . . . . .
Area exhaust port		Temperature feed water . . . . . .
Piston clearance . . .		Temperature hot well . . .
Port clearance.		Water per hour . . . .
Boiler pressure .		Coal per hour. . . . .
Initial pressure		
M.E.P..		

Indicator Card Analysis. *Meaning of Lines of Diagram.* Fig. 35 illustrates a typical indicator card with all reference lines and events of the stroke designated by letters. It is to be remembered that the indicator card shows the relation between piston position and pressures in the cylinder. So on the diagram, all vertical lines or ordinates represent pressures and all the horizontal lines or abscissas represent distances. Bearing this distinction in mind, the pressure in the engine cylinder at any piston position, measured along the horizontal line, can be obtained by measuring the vertical height of the diagram at the point representing the piston position. There

has been a set of lines placed on the indicator card shown, known as reference lines. These lines are OY, YK, and OX. The other lines DE, EF, FG, GH, etc., are drawn by the indicator, as is also the atmosphere line AB, and it is the result of one indication from one side of the engine piston, say the head end side. The diagram for the crank end would be quite similar, but reversed in position on the paper.

The reference lines are the atmosphere line AB, boiler pressure line YK, clearance line OY, and the vacuum or absolute pressure line OX. The atmosphere line is drawn by the indicator, when both sides of the indicator piston are acted upon by the pressure of the atmosphere only. Since this line is used as a reference line in meas-

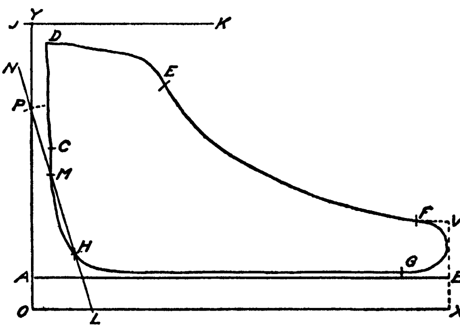


Fig 35 Typical Indicator Card

uring all pressures, it should be carefully located. The vacuum or absolute pressure line OX, the zero line of pressure, is drawn by hand below and parallel to the atmosphere line a distance by scale equal to the barometric pressure, which at sea level is 14.7 pounds per square inch.

The line of boiler pressure JK is drawn above and parallel to the atmosphere line a distance by scale equal to the boiler pressure by gauge. The distance between the boiler pressure line and the line DE represents the loss in pressure that occurs while the steam is passing from the boiler to the cylinder of the engine.

The clearance line OY is drawn at right angles to the atmosphere line and at a distance from the extremity of the diagram equal to the per cent of clearance of the engine multiplied by the horizontal length of the diagram. That is, if the clearance is 2 per cent of

the piston displacement and the length of the card is 4 inches, then the distance between the extremities of the card and the clearance line would be 2 per cent of 4, which is .08 of an inch. By the term "clearance" is meant the volumetric space between the piston and the bottom of the valve when the engine is on dead center and this may differ in amount at each end of the cylinder. Obviously this space must be filled with steam from the boiler at the initial pressure at every stroke of the piston. It is, therefore, desirable to have as small a per cent of clearance as is consistent with good design, in order to eliminate the loss of live steam. For slow-speed engines the clearance space needs to be small—about 2 to 5 per cent, whereas for high-speed engines, it may run as high as 10 per cent.

Measurement of Clearance. If the per cent of clearance is not given by the builders, it becomes necessary to measure it if any scientific study is to be made of the performance of the machine. Professor John E. Sweet gives a very simple plan for obtaining the per cent of clearance as follows:

> See that the piston and valves are made tight, and the valves disconnected. Arrange to fill the clearance space with water through the indicator holes, or through holes drilled for the purpose. Turn the engine on dead center; make marks on the crosshead and guides, weigh a pail of water, and from it fill the clearance space. Weigh the remaining water so as to determine how much is used. Then weigh out exactly the same amount of water (as is used), turn the engine off the center, pour in the second charge of water, and turn the engine back till the water comes to the same point that it did in the first instance. Make another mark on the crosshead and guide, and the distance between these marks is exactly what you really wish to know; that is, it is just what piston travel equals the clearance. If it takes one pound of water to fill this space and to admit another pound, the piston must be moved 1 inch; then the clearance bears the same relation to the capacity of the cylinder as 1 inch bears to the stroke of the piston. Thus, under these circumstances, in an engine of 10-inch stroke, it would be said to have 10 per cent clearance.

The above method would be correct when the engine is new, the piston and valves being tight and there being no leaks while the trial is being made; on the other hand, if the engine is old and the piston and valves are worn, it would only give approximate results. In such a case, it would be advisable to ascertain the per cent of clearance from the card by the following simple method. In Fig. 35 draw the straight line LN from a point L on the vacuum line in

such direction that it will cut the compression curve at two points, as H and M. Now with a pair of dividers, set one leg on the point L and adjust the other to the point H. With the dividers thus set, place one leg in the point M, where LN cuts the compression curve, then sweep an arc cutting LN at P. Erect a line perpendicular to the vacuum line and passing through the point P, which establishes the clearance line $OAPY$.

Events of the Cycle. While the steam engine is making one complete revolution, four separate and distinct events occur, namely, the admission, cut-off, release, and compression of steam. The point in the stroke where these events occur can be very accurately determined from the indicator diagram. Corresponding to the above events, there are six distinct lines on the card, namely, admission, steam, expansion, release, compression, and back pressure. By properly analyzing the diagram in Fig. 35, these events and lines are easily determined.

The admission line CD shows the rise of pressure in the cylinder due to the opening of the steam valve permitting steam to enter the cylinder. The point C indicates the point in the stroke at which the admission of steam took place

Steam line DE is drawn while the valve remains open and steam is being admitted to the cylinder.

At the point E, the valve closes the steam port, thus cutting off the supply of steam to the cylinder, hence E is the point of cut-off (c o).

As the motion continues, the volume back of the piston is increased and the pressure drops, due to the expansion of the steam, giving the expansion curve EF.

The point of release occurs at F, where the valve uncovers the exhaust port, permitting the steam to escape from the cylinder.

As soon as the point of release F is reached, the pressure begins to drop and by the time the point G has been reached on the return stroke practically all of the steam has been exhausted, hence FG is called the release line.

The back pressure line GH indicates the amount of pressure against the engine piston while making the return stroke. On noncondensing engines, it is either coincident or above the atmosphere line. On condensing engines, it is found below the atmosphere line a distance depending upon the amount of vacuum being maintained.

At the point H the exhaust port is closed and the steam remaining in the cylinder is compressed. The line HC indicates the rise of pressure due to the compression.

The events of the stroke—as cut-off at E, release at F, compression at H, and admission at C—are not always easily located on

the card, as there may not be a distinct change in the curve where these different events occur. The engineer must use his best judgment in locating these points. As an aid in making a decision, if one carefully inspects the diagram, it will be noted that when cut-off takes place at E, for instance, the steam line is concave and the expansion line convex downward, hence where these two opposite curves join must be the point at which cut-off occurs. Even this suggestion does not always hold but it will serve as an aid.

The events are usually expressed in per cents of the stroke. To obtain these per cents, proceed as follows: Locate the events as described above. Consider the point of cut-off as located on the card shown in Fig. 36. From this point, draw a line perpendicular to the atmosphere line, cutting it at the point G. Measure the length

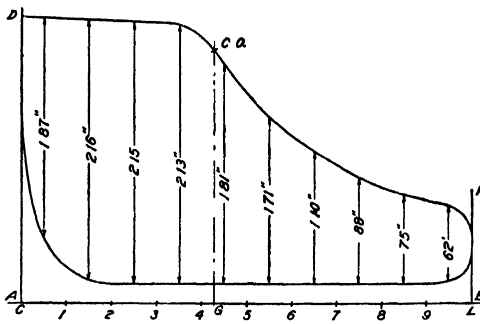


Fig 36 Indicator Card Laid Out for Determining the M E. P.

of the card between the ordinates $C\,D$ and $F\,L$; measure also the distance $C\,G$. The per cent of stroke at which cut-off occurs would be $\frac{C\,G}{C\,L} \times 100$. $C\,G$ equals 1.62 inches; $C\,L$ equals 3.72 inches. Therefore

$$\text{Per cent of cut-off} = \frac{1.62}{3.72} \times 100 = 43.5 \text{ per cent}$$

To find the per cent of release, admission, or compression, proceed in the same manner as for cut-off, always measuring the distances from the admission end of the card.

Pressures. In discussing an indicator card, five different pressures are frequently considered, namely, initial, terminal, and back pressure, pressure at the events, and the mean effective pressure

The initial pressure is the pressure in the cylinder at or near the beginning of the stroke. It would be measured on a perpendicular line from the atmosphere line to the steam line at D in Fig. 35.

The terminal pressure is the pressure measured above the vacuum line at the end of the stroke. It is the pressure that would have been acting against the piston at the end of the stroke if the steam had not been released earlier. It is measured by extending the expansion curve until it cuts a perpendicular at the end of the card at V, Fig. 35. VX measured to scale gives the amount of the terminal pressure.

Back pressure, which is the pressure the piston works against on the return stroke, is the distance between the atmosphere line AB and the back pressure line GH, Fig. 35.

The pressure at the events is obtained by scaling a perpendicular line drawn from the points in question to the atmosphere line.

The mean effective pressure, usually written m.e.p., is the net average pressure that acts on the piston throughout the entire stroke. It is evident from examination of Fig. 35 that the m.e.p. is the average height of the card multiplied by the scale of the spring used in the indicator.

There are two general ways of obtaining the m.e.p., viz, by the use of a planimeter and by the ordinate method.

In Fig. 36, CL is the atmosphere line and CD and FL are perpendiculars drawn at the end of the card. Divide CL into ten equal divisions, as *1, 2, 3*, etc., and midway between *C-1, 1-2*, etc., draw the lines shown. Measure these lines and mark their lengths on them. When this is done, obtain the sum of all of these lengths, which, in this case, is 15.18 inches; and 15.18 inches divided by 10 gives 1.518 inches, the average height of the card. If the scale of the spring used was 40 pounds, then 1.518 inches multiplied by 40 gives 60.72 pounds as the m.e.p.

The ordinate method for finding the area of a card, then, is to divide the atmosphere line into ten or more equal divisions and, half way between these divisions, erect ordinates and divide the sum of all the ordinates by the number of lines and multiply by the scale of the spring.

The average m.e.p. for one revolution would be the average of the two mean effective pressures as determined from cards taken from both the head and the crank end of the cylinder.

The number of divisions into which the card is divided could have been twenty as well as ten or any other number, but as ten or twenty makes the computations simple, they are usually taken. More accurate results will be obtained if a greater number of divisions are made, other things being equal.

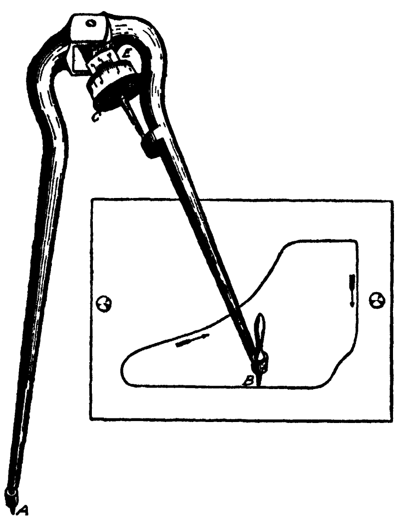


Fig 37 Measuring Area of Diagram by Means of Planimeter

Determinaton of Mean Effective Pressure by Planimeter. A more accurate way of obtaining the m.e.p. is by using an instrument called the planimeter, of which there are several types in common use. The Amsler polar planimeter is one of the most simple, and as shown in Fig. 37 is about one-half the size of the instrument. It consists of two arms free to move about a pivot and a roller graduated in square inches and tenths of square inches. A vernier is placed with the roller so the areas may be read in hundredths of a square inch. The point *A* is kept stationary and the tracer *B* is moved once around the outline of the diagram. The area in square inches of the diagram is read from the roller *C* and the vernier *E*.

INSTRUCTIONS FOR USE OF PLANIMETER

The diagram should be fastened to some flat unglazed surface, such as a drawing board, by means of thumb tacks, springs, or pins. The point *A* is pressed into the paper so that it will hold in place, *B* is set at any point in the outline of the diagram, and the roller is set at zero. Follow the outline of the diagram carefully in the direction of the hands of a watch, as indicated by the arrows in Fig. 37, until the tracer has moved completely around the diagram The result is then read to hundredths of an inch from the roller. Suppose, after tracing over the outline, we find that the largest figure that has passed the zero of the vernier is 3; the number of graduations (tenths) that have passed the zero are 5; and the graduation (hundredths) on the vernier that exactly coincides with a graduation on the roller is 9. Then the area is 3.59 square inches.

Often at the start the roller is not adjusted so that the zeros coincide, but the reading is taken and subtracted from the final reading. Thus, if the

first reading is 4.63, and the second 7.31, the area is 7.31 — 4.63 or, 2.68 square inches. In case the second reading is less than the first, add 10 to the second reading, then subtract.

Briefly stated, to find the m.e p. of a diagram, first ascertain the area of the card by use of the planimeter, multiply the area obtained by the scale of the spring, and divide the product by the length of the card in inches. This may be demonstrated in the following manner: If in a rectangle A equals area in square inches, L equals length in inches, and H equals height in inches, then

$$A = L H$$

If P is the average pressure, then

$$H = \frac{\text{average pressure}}{\text{scale of spring}}$$

or

$$H = \frac{P}{\text{scale}}$$

Substituting this value in the equation $A = L\ H$, we have

$$A = \frac{L \times P}{\text{scale}}$$

or

$$P = \frac{A \times \text{scale}}{L}$$

The planimeter is a very valuable instrument to an engineer in taking indicator cards, the results obtained being very accurate. Ten or twelve diagrams can be measured by this instrument in the same time that is necessary to measure a single card by the method of ordinates.

It is well to run over the area two or three times and take an average, as the tracing of the diagram can not be absolutely correct at any time.

PHYSICAL THEORY

In the study of any subject, there are always a number of technical terms that need to be clearly understood before a proper understanding of the subject is obtained. This is especially true with the steam engine, and the indicator, and with steam which is the motive force of each. It is, therefore, thought best to treat these terms and a study of the properties of steam together, at this point, in order to be able to go more deeply into the study of the uses of the indicator.

Pressure. Pressure is the force tending to compress a body and is usually expressed either in pounds per square inch, pounds per square foot, inches of mercury, or inches of water.

Boiler Pressure. Boiler pressure is the pressure of steam in pounds per square inch above atmosphere as indicated by a steam gauge.

Absolute Pressure. Absolute pressure is the pressure as obtained from absolute vacuum. At the sea level, the pressure of the atmosphere is usually taken as 14.7 pounds per square inch; hence, if at sea level, a steam gauge reads one hundred pounds, the absolute pressure would be $100+14.7$ or 114.7 pounds per square inch absolute.

Atmospheric Pressure. Atmospheric pressure is the pressure the atmosphere exerts upon a body. It is usually measured in inches of mercury, as obtained from a barometer, hence it is sometimes spoken of as barometer pressure. Since the weight of 1 cubic inch of mercury is known to be .49 pounds, the reading of a barometer can be easily converted into pounds per square inch. If a barometer reads 28 inches of mercury, the pressure of the atmosphere expressed in pounds per square inch would be $28\times.49=13.72$.

Pressure below atmosphere is also given in inches of mercury and sometimes in inches of water. If it is desired to obtain the absolute pressure in pounds per square inch, reduce the reading in inches of mercury to pounds per square inch and subtract this amount from the atmospheric pressure expressed in pounds per square inch.

EXAMPLE. A vacuum gauge on a steam engine condenser reads 26 1 inches of mercury. The barometer stands at 29.12 inches of mercury. What is the absolute pressure in the condenser in pounds per square inch?

SOLUTION. 29 12 inches of mercury is equivalent to $29.12\times.49=14.268$ pounds per square inch and 26.1 inches is equivalent to $26.1\times.49=12.789$ pounds per square inch. Therefore, the absolute pressure in pounds per square inch in the condenser is $14.268-12.789=1.479$.

Work. The unit of work is called a *foot pound* and it is equal in amount to the energy required to lift one pound one foot high. It is to be noted that it is the product of a force times the distance through which it acts. If a weight of 50 pounds be raised 7 feet high, then 50×7 or 350 foot pounds of work would be expended.

Heat. *Temperature.* By temperature is meant simply the thermal condition of a body with reference to its capability of transferring heat to other bodies. If two bodies are placed in contact and the first gives more heat to the second than it receives, we say that No. 1 is hotter than No. 2. If the first receives more heat than it gives, No. 2 is hotter than No. 1. If both bodies give and receive the same amount of heat, they are of the same temperature.

According to our theory, it is evident that temperature depends upon the energy of molecular vibration. If the temperature rises, it means that the molecular vibration, and consequently the energy increases; if the temperature falls, the energy of molecular vibration decreases. Evidently, a point must finally be reached when this energy of vibration is zero and the molecules are at rest. At this temperature, there is no heat and we call it the absolute zero. This zero is evidently must below the zero of the ordinary scale.

Thermometers. In order to determine just how hot a body is, we must compare its temperature with that of some substance whose degree of heat we know. As it would be impossible to keep several bodies at different degrees of heat for comparison, we must resort to some other means. A simple method is to use some substance whose volume changes a definite amount for a definite change in temperature and always has the same volume for the same temperature. Mercury and alcohol are suitable substances and may be placed in a glass bulb, to which is connected a glass tube of small bore. All the air is drawn out of the tube, and the end is sealed so that the thermometric substance can expand or contract in a vacuum. The tube having been sealed, the bulb is placed in melting ice and the height of the mercury in the tube noted. It is then placed in steam (or boiling water) at atmospheric pressure and the height of the column again noted. On the Fahrenheit scale, the melting point is called 32 degrees and the boiling point 212 degrees, and the intervening space is divided into 180 equal parts. In the centigrade scale, the melting point is called zero degree and the boiling point 100 degrees; there are 100 equal intervals between them. Thus we see that $180°F. = 100°C.$, or $1°C. = 1.8°F.$

Example What is the temperature of 50°C. on the Fahrenheit scale?

$$50°C. = 50 \times 1.8 = 90°F. \text{ above the melting point}$$
$$= 90 + 32 = 122°F.$$

In order to compare temperatures, we place the thermometer in contact with the substance whose degree of heat we wish to know and then observe the height of the liquid column in the thermometer. The height of this column depends upon the expansion of the thermometric substance and indicates the intensity of heat, or the temperature as we commonly call it. We use a thermometer to measure the *intensity* of heat, but not the *quantity* of heat.

Unit of Heat Quantity. For measuring the intensity of heat, the degree is the unit; for measuring the quantity of heat, we have another unit, which is the amount of heat necessary to raise one pound of water from 61° F. to 62° F. This is called the British thermal unit (B.T.U.). To raise one pound of water from 60° F. to 62° F., or to raise two pounds from 60° F. to 61° F., will require 2 B.T.U.

One B.T.U. is equivalent to 778 foot pounds of work. If one pound of coal liberates 12,000 B.T.U. when burned, it is capable of producing $12{,}000 \times 778 = 9{,}336{,}000$ foot pounds of work.

The above value of the B.T.U. in foot pounds of work is known as the mechanical equivalent of heat, that is, 778 foot pounds.

Horsepower. Horsepower is the arbitrary standard used for measuring the power of a steam engine. It is said to have been originally established by James Watt from experiments conducted with dray horses on the streets of London. Its value is, however, considerably above that of the ordinary horse. It is defined as being equal to lifting 33,000 pounds one foot high in one minute of time. It will be noted that the horsepower takes into account the following factors: *force*, *distance*, and *time*. This being true, it is desirable to have an expression in the form of an equation to express the horsepower of an engine. The common formula for steam engines is

$$\text{h.p.} = \frac{PLAN}{33000}$$

in which P equals the mean effective pressure in pounds per square inch; L equals length of stroke in feet; A equals area of cylinder in square inches; N equals number of revolutions per minute (r.p.m.); and 33,000 equals equivalent foot pounds of work per minute in one horsepower.

Analyzing the equation, it is found that it conforms to the defini-

tion of work previously given. For instance, A, the area of the piston in square inches, times P, the mean effective pressure in pounds per square inch, is equal to the total force on the piston which acts through the stroke a distance of L feet. Hence, the expression $P L A$ gives the foot pounds of work done during one stroke. In the definition of horsepower, it was noted that the time element was considered, so we have $(P L A) \times N$ divided by 33,000, fulfilling the definition of a horsepower, since N involves the element of time.

Indicated Horsepower. The indicated horsepower (i.h.p.) is the computed horsepower of an engine as obtained by using an indicator diagram taken from the engine cylinder. From this diagram is determined P, the mean effective pressure, which is substituted in the equation just given.

EXAMPLE. Given an engine having a cylinder 10 inches in diameter and a stroke 16 inches in length, running at 180 r.p.m. The mean effective pressure on the piston as obtained from the indicator card is 75 pounds on both the head and the crank end of the cylinders. Required the horsepower of the engine.

SOLUTION. In this example P equals 75 pounds; L equals $16 \div 12$, or 1.33 feet; A equals πR^2 equals 3.1416×5^2 or 78.54 square inches; and N equals 180 r.p m.

Substituting these values in the formula

$$\text{h.p.} = \frac{P L A N}{33000}$$

we have

$$\text{h.p.} = \frac{75 \times 1.33 \times 78.54 \times 180}{33000} = 42.75$$

This is for one end of the cylinder only. For both ends, we get the total

$$\text{h.p.} = 42.75 \times 2 = 85.5 \text{ (approximately)}$$

ENGINE CONSTANT. It is evident from the foregoing problem that, for a given engine, some factors in the h.p. formula remain constant. These constants are: the area of the piston, the length of the stroke, and the abstract number 33,000. It is convenient when making a large number of computations to determine what is known as the engine constant, a factor which saves considerable time and reduces the chances of error. Since the area of the piston on the crank end is smaller than that on the head end by an amount equal to the area of the piston rod, the engine constant for the crank end is always slightly smaller than for the head end.

EXAMPLE Find the constant for both h e. and c e. of the engine in the preceding problem, whose piston rod was 1¾ inches in diameter. The engine constant for the head end is

$$C_{h\ e.} = \frac{L\ A}{33000}$$

in which L equals 16÷12, or 1.33 feet, A equals 3.1416×5^2 or 78.54 square inches.

$$\therefore \qquad C_{h.e.} = \frac{1.33 \times 78\ 54}{33000} = .00316$$

For the crank end, the area of piston is reduced by the area of the 1¾-inch piston rod, which area is equal to 2.40 square inches. The effective area for the crank end is, therefore, 78.54 − 2.40, or 76.14 square inches.

$$\therefore \qquad C_{c.e.} = \frac{1.33 \times 76.14}{33000} = 003068$$

Having obtained the engine constant, in order to obtain the indicated horsepower (i h p.) it is only necessary to multiply the m e p. on the piston for each end of the cylinder by the engine constant for that end and by the number of revolutions

In Table II, the approximate i.h.p. of an engine is easily found by multiplying the constant, corresponding to the diameter of the piston, by the piston speed and by the m.e.p. Or, in other words, the constants in the table equal the h.p. for an engine with a given diameter of piston having a piston speed of one foot per minute and a m.e.p. of one pound. The piston speed of any engine is equal to the length of stroke in feet multiplied by twice the number of revolutions. For instance, in the 10- by 16-inch engine already referred to, the piston speed in feet per minute would be 1.35 feet × 180 × 2 = 478.8 feet per minute.

If the diameter of the piston is an even number, the constant is found in the second column; if it contains a fraction, the constant is found by following the column horizontally until the required fraction is reached. The constant multiplied by the piston speed in feet per minute and by the m.e.p. in pounds per square inch gives the i.h.p. approximately.

EXAMPLE An engine runs at 75 r p.m and the stroke is 4 feet. If the m.e.p. is 48 and the piston is 27¾ inches in diameter, determine the i.h.p.

SOLUTION. From Table II, the constant for a piston 27¾ inches in diameter is .0178355 The piston speed is 4 × 75 × 2 = 600 feet per minute Then

$$\text{i.h p.} = 0178355 \times 600 \times 48$$
$$= 513.66 \text{ (approx.)}$$

TABLE II

Engine Constants

Diameter of Cylinder	Even Inches	+⅛″ or .125	+¼″ or 25	+⅜″ or 375	+½″ or 5	+⅝″ or 625	+¾″ or 75	+⅞″ or 875
1	0000238	0000301	0000372	0000450	0000535	0000628	0000729	0000837
2	0000952	0001074	0001205	0001342	0001487	0001640	0001800	0001967
3	0002142	0002324	0002514	0002711	0002915	0003127	0003347	0003574
4	0003808	0004050	0004299	0004554	0004819	0005091	0005370	0005656
5	0005950	0006251	0006560	0006876	0007199	0007530	0007869	0008215
6	0008568	0008929	0009297	0009672	0010055	0010445	0010844	0011249
7	0011662	0012082	0012510	0012944	0013387	0013837	0014295	0014759
8	0015232	0015711	0016198	0016693	0017195	0017705	0018222	0018746
9	0019278	0019817	0020363	0020916	0021479	0022048	0022625	0023209
10	0023800	0024398	0025004	0025618	0026239	0026867	0027502	0028147
11	0028798	0029456	0030121	0030794	0031475	0032163	0032859	0033561
12	0034272	0034990	0035714	0036447	0037187	0037934	0038690	0039452
13	0040222	0040999	0041783	0042576	0043375	.0044182	0044997	0045819
14	0046648	0047484	0048328	0049181	0050039	0050906	0051780	0052661
15	0053550	0054446	0055349	0056261	0057179	0058105	0059039	0059979
16	0060928	0061884	0062847	0063817	0064795	0065780	0066774	0067774
17	0068782	0069797	0070819	0071850	0072887	.0073932	0074985	0076044
18	0077112	0078187	0079268	0080360	0081452	.0082560	0083672	0084791
19	0085918	0087052	0088193	0089343	0090499	0091663	0092835	0094013
20	0095200	0096393	0097594	0098803	0100019	0101243	0102474	0103712
21	0104958	0106211	0107472	0108739	0110015	0111299	0112589	0113886
22	0115192	0116505	0117825	0119152	0120487	0121830	0123179	0124537
23	0125902	0127274	0128654	0130040	0131435	0132837	0134247	0135664
24	0137088	0138519	0139959	0141405	0142859	0144321	0145789	0147266
25	0148750	0150241	0151739	0153246	0154759	0156280	0157809	0159345
26	0160888	0162439	0163997	0165563	0167135	0168716	0170304	0171899
27	0173502	0175112	0176729	0178355	0179988	0181627	0183275	0184929
28	0186592	0188262	0189939	0191624	0193316	0195015	0196722	0198436
29	0200158	0201887	0203624	0205368	0207119	0208879	0210645	0212418
30	0214200	0215988	0217785	0219588	0221399	0223218	0225044	0226877
31	0228718	0230566	0232422	0234285	0236155	0238033	0239919	0241812
32	0243712	0245619	0247535	0249457	0251387	0253325	0255269	0257222
33	0259182	0261149	0263124	0265106	0267095	0269092	0271097	0273109
34	0275128	0277155	0279189	0281231	0283279	0285336	0287399	0289471
35	0291550	0293636	0295729	0297831	0299939	0302056	0304179	0306309
36	0308448	0310594	0312747	0314908	0317075	0319251	0321434	0323624
37	0325822	0328027	0330239	0332460	0334687	0336922	0339165	0341415
38	0343672	0345937	0348209	0350489	0352775	0355070	0357372	0359681
39	0361998	0364322	0366654	0368993	0371339	0373694	0376055	0378424
40	0380800	0383184	0385575	0387973	0390379	0392793	0395214	0397642
41	0400078	0402521	0404972	0407430	0409895	0412368	0414849	0417337
42	0419832	0422335	0424845	0427362	0429887	0432420	0434959	0437507
43	0440062	0442624	0445194	0447771	0450355	0452947	0455547	0458154
44	0460768	0463389	0466019	0468655	0471299	0473951	0476609	0479276
45	0481950	0484631	0487320	0490016	0492719	0495430	0498149	0508875
46	0503608	0506349	0509097	0511853	0514615	0517386	0520164	0522949
47	0525742	0528542	0531349	0534165	0536988	0539818	0542655	0545499
48	0548352	0551212	0554079	0556953	0559835	0562725	0565622	0568526
49	0571438	0574357	0577284	0580218	0583159	0586109	0589065	0592029
50	0595000	0597979	0600965	0603959	0606959	0609969	0612984	0616007
51	0619038	0622076	0625122	0628175	0632235	0634304	0637379	0640462
52	0643552	0646649	0649753	0652867	0655987	0659115	0662250	0665392
53	0668542	0671699	0674864	0678036	0681215	0684402	0687597	0690799
54	0694008	0697225	0700449	0703681	0705293	0710166	0713419	0716681
55	0719950	0724226	0726510	0729801	0733099	0736406	0739719	0743039
56	0746368	0749704	0753047	0756398	0759755	0763120	0766494	0769874
57	0773262	0776657	0780060	0783476	0786887	0790312	0793745	0797185
58	0800632	0804087	0807549	0811019	0814495	0817980	0821472	0824971
59	0828478	0831992	0835514	0839043	0842579	0846123	0849675	0853234
60	.0856800	0860374	0863955	.0867543	0871139	0874743	.0878354	.0881973

The result is only approximately correct on account of the area of the piston rod not being deducted from the area of the piston on the crank end. It is sufficiently accurate, however, for practical purposes.

Brake Horsepower. All of the i.h.p. is not available for useful work as the internal friction of the engine absorbs some of the energy, so the net horsepower is the i.h.p. less the h.p. consumed by the engine in overcoming internal resistances. This net horsepower is usually spoken of as the brake horsepower (b.h.p.) and it is obtained by the use of some form of brake.

Mechanical Efficiency. The mechanical efficiency of an engine is the ratio between the b.h.p. and the i.h.p. Expressed in per cent, it would be $\frac{\text{b.h.p.} \times 100}{\text{i.h.p.}}$. It is sometimes given as the engine friction in per cent, that is, the mechanical efficiency is expressed as ten or fifteen per cent engine friction, which is evidently 100 minus the mechanical efficiency. Under ordinary conditions, the engine friction varies from about 6 to 10 per cent of the i.h.p. depending on the size and the construction of the engine.

Piston Displacement. The piston displacement is the space in the cylinder swept through by the piston in its travel, expressed in cubic feet. The piston displacement for the c.e. will be less than for the h.e. by an amount equal to the area of the piston rod in square feet multiplied by the stroke in feet. In the 10- by 16-inch engine with $1\frac{3}{4}$-inch piston rod, the piston displacement for head end is

$$\underset{\text{(Head End)}}{\text{Piston displacement}} = \frac{.7854 \times 10^2 \times 16}{1728}$$
$$= .72722 \text{ cubic feet}$$

For the crank end, it would be .72722 less the cubic feet in the piston rod or

$$\underset{\text{(Crank End)}}{\text{Piston displacement}} = .72722 - \left(\frac{.7854 \times 1.75^2 \times 16}{1728}\right)$$
$$= .72722 - .02221$$
$$= .70501 \text{ cubic feet}$$

PROPERTIES OF STEAM

Saturated Vapor. The process of converting a liquid into a vapor is known as vaporization; the product thus formed is readily condensed and, therefore, does not follow the laws of perfect gases. A dry saturated vapor is one that has just enough heat in it to keep

it in the form of a vapor; if we add more heat, it becomes superheated. A superheated vapor may lose a part of its heat without condensation; a saturated vapor can not. When a saturated vapor loses a part of its heat, some of it will condense and we say that the vapor is wet.

Steam is simply the vapor from water and we shall confine our discussion to this alone. Suppose we have a vertical cylinder, as shown in Fig. 38, fitted with a light piston free to move up and down, yet so constructed that it may be loaded at will. Suppose that there is one pound of water at a temperature of 32°F. in the bottom of cylinder A, and that the piston rests upon its surface. Now, if heat is applied by means of a gas flame or fire, we shall notice the following effects:

(1) The temperature of the water will gradually rise until it reaches the temperature at which steam is formed. This temperature will depend upon the pressure, or the load on the piston. If the piston is very light, we may neglect its weight and consider that there is simply the atmospheric pressure of 14.7 pounds per square inch acting on the water surface, at which pressure, steam forms at 212° F.

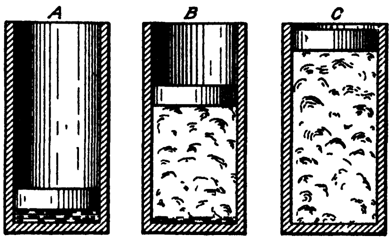


Fig 38 Engine Cylinder Containing Water and Steam

(2) Therefore, as soon as 212 degrees is reached, steam will form and the piston will steadily rise (B, Fig. 38), but no matter how hot the fire may be, the temperature of both water and steam will remain at 212 degrees until all the water is evaporated (C, Fig. 38).

We had one pound of water at 32 degrees and at 14.7 pounds absolute pressure, and found that steam formed at a temperature of 212 degrees and remained at that temperature. There were added 180.3 B.T.U., the heat of the liquid, to bring the water from 32 degrees to the boiling point. To convert water at 212 degrees into steam at 212 degrees, there were added 969.7 B.T.U. more. This quantity, known as the latent heat, or heat of vaporization, makes the total heat 1150.0 B.T.U. If we should measure the volume carefully after all the water was evaporated, we should find

that there were exactly 26.78 cubic feet of dry saturated steam. At the start we had one pound of water and, therefore, we must have one pound of steam, for none could escape; hence one cubic foot will weigh $\frac{1}{26.78}$, or 0.03734 pound, which is known as the density of steam at 14.7 pounds absolute pressure and 212° F.

Effect of Pressure on Boiling Point. Suppose now we place a weight of 85.3 pounds per square inch on the piston. The pressure is 85.3 plus 14.7 or 100 pounds per square inch absolute. We shall now find that no steam will form until a temperature of 327.86 degrees is reached, and also we must add 887.6 B.T.U. Under this greater pressure the steam occupies a volume of only 4.432 cubic feet, or one cubic foot of it weighs $\frac{1}{4.432}$, or 0.2256 pound.

From the foregoing, it is obvious that there is a definite relation between the pressure and the temperature, that is, the temperature at which water will boil depends upon the pressure, and *vice versa.*

Of course, it would be impossible to determine all these different quantities by actual experiment, and at all pressures varying from vacuum to the high pressures used in water-tube boilers; they can be computed.

Steam Tables. We have already seen that any change in the temperature of saturated steam produces a change of pressure, and that every change of pressure corresponds to a certain change in temperature. There are several properties of saturated steam that depend upon the temperature and pressure; and the values of all these different properties when arranged for all temperatures and pressures are called Steam Tables. The following are the principal items found in the tables:

(1) *Absolute pressure* in pounds per square inch; it is equal to the gauge pressure plus the atmosphere pressure of 14.7 pounds or the pressure of atmosphere as obtained from the barometer.

(2) *Temperature of steam,* or of boiling water, at the corresponding pressure.

(3) *Heat of liquid,* or the number of B.T.U. necessary to raise one pound of water from 32°F. to the boiling point corresponding to the given pressure.

(4) *Heat of vaporization,* or latent heat; that is, the number of B.T.U. necessary to change one pound of water, at the boiling point, into dry satur-

ated steam at the same temperature and pressure. It was noted that in heating water from 32°F. to the boiling point under atmosphere pressure, the temperature rose from 32 degrees to 212 degrees, but as soon as 212 degrees was reached, the temperature remained constant until all the water was converted into steam at that pressure While the process of changing the water at 212 degrees into steam at 212 degrees was going on, heat was being added but no change occurred in temperature. This phenomenon always occurs when the application of heat results in the change of state of a substance, either from a solid to a liquid, or from a liquid to a gaseous state This apparent loss of heat is not real, but recurs whenever the transformation is reversed, that is when the steam is condensed

(5) *Total heat of vaporization* is the number of B.T.U necessary to change one pound of water from 32°F. into steam at the given temperature or pressure, and represents the sum of the heat units of the liquid plus the heat units of vaporization or the latent heat

(6) *Density of steam*, which is the weight of one cubic foot of steam at the given temperature or pressure

(7) *Specific volume*, which is the volume occupied by one pound of steam The specific volume is the reciprocal of the density, that is, it is equal to $\frac{1}{\text{density}}$.

Steam tables from which the above items may be obtained are very useful to the engineer. Any one wishing to make a more detailed study of the steam tables and the properties of steam should procure "Steam and Entropy Tables" by Peabody, published by John Wiley & Sons.

Kinds of Steam. If the process of adding heat to water and then to steam be continued, three kinds of steam are obtained depending on the conditions, namely, saturated or dry steam, wet steam, and superheated steam.

Saturated or Dry Steam. If just sufficient heat be added to the vessel A, Fig. 38, until the pound of water is completely converted into steam as shown in C, the result is a pound of saturated steam. The number of heat units that have been added equals the heat of the liquid q plus the latent heat r. Therefore Q, the number of B.T.U. added, in equation form becomes

$$Q = q + r$$

It is evident, therefore, that saturated steam contains all the heat of the liquid plus the heat of vaporization.

Wet Steam. If instead of adding enough heat to the vessel C to completely evaporate the water, the operation be discontinued while there is yet some water remaining unevaporated, as is shown

TABLE III
Properties of Saturated Steam

Total pressure in lbs per sq in above vacuum p	Temperature in degrees Fahrenheit t	Heat in liquid from 32° in heat units q	Heat of vaporization or latent heat in heat units r	Total heat in heat units from water at 32° H	Density or weight of one cubic foot in lbs $\frac{1}{s}$	Volume of 1 pound in cubic feet s	Total pressure in lbs per sq in above vacuum p
1	101 84	69 8	1034 7	1104 5	0 00300	333 1	1
2	126 15	94 2	1021 9	1116 1	0 00578	173 1	2
3	141 52	109 6	1012 2	1121 8	0 00845	118 4	3
4	153 00	121 0	1005 5	1126 5	0 01106	90.4	4
5	162 26	130 3	1000 0	1130 3	0 01364	73 3	5
6	170 07	138 1	995 5	1133 6	0 01616	61 9	6
7	176 84	144 9	991 4	1136 3	0 01866	53 6	7
8	182 86	150 9	987 8	1138 7	0 02116	47 26	8
9	188 27	156 4	984 5	1140 9	0 02362	42 36	9
10	193 21	161 3	981 4	1142 7	0 02606	38 37	10
14 7	212 00	180 3	969 7	1150 0	0 03734	26 78	14 7
15	213 03	181 3	969 1	1150 4	0 03805	26 28	15
20	227 95	196 4	959 4	1155 8	0 04978	20 09	20
25	240 07	208 7	951 4	1160 1	0 06140	16 29	25
30	250 34	219 1	944 4	1163 5	0 0728	13 74	30
35	259 29	228 2	938 2	1166 4	0 0842	11 88	35
40	267 26	236 4	932 6	1169 0	0 0953	10 49	40
45	274 46	243 7	927 5	1171 2	0 1065	9 387	45
50	281.03	250 4	922 8	1173 2	0 1176	8 507	50
55	287 09	256 6	918 4	1175 0	0 1286	7 778	55
60	292 74	262 4	914 3	1176 7	0 1395	7 166	60
65	298 00	267 8	910 4	1178 2	0 1504	6 647	65
70	302 96	272 9	906 6	1179 5	0 1613	6 199	70
75	307.64	277 7	903 1	1180 8	0 1722	5 807	75
80	312 08	282 2	899 8	1182 0	0 1829	5 466	80
85	316 30	286 5	896 6	1183 1	0 1938	5 161	85
90	320 32	290 7	893 5	1184 2	0 2047	4 886	90
95	324 16	294.6	890 5	1185 1	0 2153	4 644	95
100	327 86	298 5	887 6	1186 1	0 2256	4 432	100
105	331 42	302 1	884 8	1186 9	0 2362	4 233	105
110	334 83	305 6	882 1	1187 7	0 2471	4 047	110
115	338 14	309 0	879 5	1188 5	0 2580	3 876	115
120	341 31	312 3	876 9	1189 2	0 2686	3 723	120
125	344 39	315 5	874 5	1190 0	0 2793	3 581	125
130	347 38	318 6	872 1	1190 7	0 2898	3 451	130
140	353 09	324 4	867 4	1191 8	0 3106	3 220	140
150	358 50	330 0	863 0	1193 0	0 3318	3 014	150
160	363 62	335 3	858 8	1194 1	0 3528	2 834	160
170	368 50	340 4	854 8	1195 2	0 3741	2 673	170
180	373 16	345 2	850 9	1196 1	0 3951	2 531	180
190	377 61	349 8	847 1	1196 9	0 4158	2 405	190
200	381 89	354 3	843 5	1197 8	0 4371	2 288	200
210	386 02	358 6	840 0	1198 6	0 4579	2 184	210
220	389 98	362 7	836 6	1199 3	0 4789	2 088	220
230	393 80	366 6	833 3	1199 9	0 4997	2 001	230
240	397 50	370 5	830 1	1200 6	0 521	1 921	240
250	401 10	374 2	826 9	1201 1	0 542	1 845	250
260	404 55	377.8	823 9	1201.7	0 563	1 775	260
270	407 90	381 3	820 9	1202.2	0 584	1 711	270
280	411 19	384 8	818 0	1202.8	0 605	1 652	280
290	414 35	388 1	815 2	1203 3	0 627	1 595	290
300	417 45	391.3	812 4	1203 7	0 649	1 542	300
310	420 45	394.4	809 7	1204 1	0 670	1.492	310
320	423 40	397 5	807 1	1204 6	0.692	1 446	320

TABLE IV
Properties of Saturated Steam

Temperature in degrees Fahrenheit t	Total pressure above vacuum p	Heat in liquid in heat units from 32° q	Heat of vaporization or latent heat in heat units r	Total heat in heat units from water at 32° H	Density or weight of one cubic foot in lbs $\frac{1}{s}$	Volume of 1 pound in cubic feet s	Temperature in degrees Fahrenheit t
32	0 0886	0 0	1071 7	1071 7	0 000302	3308 0	32
60	0 2561	28 1	1057 0	1085 1	0 000828	1207 0	60
90	0 6960	58 1	1041 2	1099 3	0 002131	469 2	90
120	1 689	88 0	1024 4	1112 4	0 004926	203	120
140	2 885	108 0	1013 1	1121 1	0 00814	122 8	140
150	3 715	118 0	1007 2	1125 2	0 01032	96 9	150
160	4 738	128 0	1001 4	1129 4	0 01296	77 2	160
170	5 990	138 0	995 5	1133 5	0 01613	62 0	170
180	7 510	148 0	989 5	1137 5	0 01993	50 2	180
190	9 339	158 1	983 4	1141 5	0 02444	40 92	190
200	11 528	168 2	977 2	1145 4	0 02974	33 62	200
212	14 698	180 3	969 7	1150 0	0 03734	26 78	212
220	17 188	188 4	964 6	1153 0	0 04321	23 14	220
225	18 914	193 4	961 4	1154 8	0 04726	21.16	225
230	20 780	198 5	958 1	1156 6	0 05160	19 37	230
235	22 790	203 6	954 8	1158 4	0 05630	17.77	235
240	24 970	208 6	951 4	1160 0	0 06130	16 31	240
245	27 310	213 7	948 1	1161 8	0 06660	15 01	245
250	29 82	218 8	944 7	1163 5	0 0724	13.82	250
255	32 53	223 8	941 2	1165 0	0 0785	12.73	255
260	35 42	229 0	937 8	1166 8	0 0851	11 75	260
265	38 53	234 0	934 3	1168 3	0 0920	10 87	265
270	41 84	239 1	930 7	1169 8	0 0995	10 05	270
275	45 39	244 2	927 2	1171 4	0 1074	9 309	275
280	49 19	249 4	923 6	1173 0	0 1158	8 639	280
285	53 22	254 5	920 0	1174 5	0 1247	8 021	285
290	57 53	259 6	916 3	1175 9	0 1341	7 454	290
295	62 11	264 7	912 6	1177 3	0 1441	6 937	295
300	66 98	269 8	908 9	1178 7	0 1547	6 462	300
305	72 15	274 9	905 1	1180 0	0 1660	6 024	305
310	77 63	280 1	901 3	1181 4	0 1779	5 622	310
315	83 44	285 2	897 6	1182 8	0 1903	5 254	315
320	89 59	290 4	893 7	1184 1	0 2038	4 907	320
325	96 12	295 5	889 8	1185 3	0 2177	4 594	325
330	102 98	300 6	885 9	1186 5	0 2319	4 312	330
335	110 25	305 8	882 0	1187 8	0 2476	4 038	335
340	117 91	310 9	878 0	1188 9	0 2642	3 784	340
345	126 00	316 1	874 0	1190 1	0 2813	3 554	345
350	134 52	321 3	870 0	1191 3	0 2992	3 342	350
355	143 46	326 4	865 9	1192 3	0 3178	3 147	355
360	152 89	331 6	861 8	1193 4	0 3378	2 960	360
365	162.77	336 8	857 7	1194 5	0 3588	2 787	365
370	173 17	341 9	853 5	1195 4	0 3808	2 626	370
375	184 08	347.1	849 3	1196 4	0 4035	2 478	375
380	195 52	352.3	845 1	1197 4	0 4275	2 339	380
385	207 49	357.5	840 8	1198 3	0 4527	2 209	385
390	220 05	362 7	836 6	1199 3	0 4789	2.088	390
395	233 20	367 9	832 2	1200 1	0 5060	1 975	395
400	246 9	373 1	827 9	1201 0	0 535	1.868	400
405	261.3	378 3	823 5	1201 8	0 566	1 766	405
410	276.3	383.5	819 1	1202 6	0 598	1 673	410
415	292 0	388 7	814 6	1203.3	0 631	1 584	415
420	308.5	394 0	810 1	1204 1	0.667	1 499	420
425	325 6	399.2	805.6	1204 8	0 704	1.421	425

in the bottom of *B*, the result would be steam with some water in suspension. Steam in this state is known as wet steam, that is, it contains some moisture. Expressed in equation form, the number of B.T.U. added would be

$$Q=q+xr$$

in which x = the weight of the part vaporized. By means of a steam calorimeter, the amount of water held in suspension by the steam may be determined. In practice, it amounts to about one to three per cent, depending upon the mechanical construction of the plant and will average about two per cent. The quality of steam, considering saturated steam as unity or one, would then be $100-2=98$ per cent, dry. So the quantity x is the 98 per cent or the per cent of the total amount of water that has been vaporized.

Superheated Steam. If after saturated or dry steam is obtained, additional heat be added by some means until the temperature of the dry steam is above that corresponding to the pressure, it is said to be superheated. In obtaining superheated steam, more B.T.U. have been added than when dry steam was obtained, so another expression is used to represent the total heat B.T.U. added, viz,

$$Q=q+r+.48\ (t_s-t)$$

in which t_s equals the temperature of the superheated steam in degrees F. and is obtained by the use of thermometers; t equals the temperature corresponding to the absolute boiler pressure; and .48 equals the specific heat of superheated steam at constant pressure. This factor varies slightly for different pressures and temperatures, but for general use the value given is sufficiently accurate. It may be obtained at any pressure and temperature by experiment.

It is obvious from the foregoing that the number of B.T.U. contained in one or more pounds of steam, be it wet, dry, or superheated, can be obtained by the use of the above formulas and the steam tables.

In order to become familiar with the above formulas and the use of the steam tables, a few simple problems will be worked out. It must be borne in mind in the use of these tables that whenever a pressure is given, the other properties, such as t, q, r, etc., are found in Table III; and that if the temperature be given, Table IV must

be used. It is also to be remembered that the tables are based on absolute pressures, so if the gauge pressure be given instead of the absolute pressure, the gauge pressure reading must be converted into absolute pressure by adding the atmospheric pressure. For example, if 160 pounds gauge pressure, say at sea level, is given instead of 160 pounds absolute, then before looking for t, r, etc., corresponding to that pressure, 14.7 pounds should be added to the 160 pounds, making the absolute pressure 174.7 pounds. If the barometric pressure is 29.4 inches of mercury when the gauge pressure is 160 pounds, then the absolute pressure will be $160+(29.4\times.49)=160+14.4$, or 174.4 pounds per square inch. This must always be done before making use of the steam tables.

ILLUSTRATIVE PROBLEMS

EXAMPLE 1 How many heat units in one pound of water at 160°F ?

SOLUTION. Looking down the first column of Table IV until 160 degrees is found, then following across horizontally, we find in the third column under Heat of the Liquid 128 0 B T U , which is the number of heat units contained in one pound of water at 160°F

EXAMPLE 2. What temperature corresponds to 160 pounds absolute?

SOLUTION. Since the tables are based on absolute pressures and the pressure of 160 pounds is given as absolute, we turn to Table III and follow down the first column until 160 pounds is reached, then horizontally across to the second column where we find the temperature corresponding to 160 pounds absolute to be 363 62°F.

EXAMPLE 3. What is the heat of vaporization r at 160°F ?

SOLUTION Since it is temperature that is given, it is necessary to find 160 degrees in the first column of Table IV and following across horizontally to the fourth column, we find that the heat of vaporization r is 1001 4 B.T U.

EXAMPLE 4. What is the value of r for 160 pounds absolute pressure?

SOLUTION Since the pressure is given, it is necessary to look in the first column of Table III for 160 pounds, and following across to the fourth column we find r to be 858 8 B T U.

EXAMPLE 5. Steam is made in a boiler at 140 pounds per square inch absolute from feed water at a temperature of 70°F , 99 per cent of each pound being evaporated. How many heat units are spent in raising the temperature of one pound of the water to the boiling point? What are the total number of B.T U required to make one pound of steam?

SOLUTION. Looking for q in Table IV, corresponding to the temperature of 70°F , we find that it is necessary to interpolate between 90 degrees and 60 degrees. For 90 degrees, q is 58 1, for 60 degrees it is 28 1 The difference is $58\ 1-28\ 1$, or 30 0 for a difference of 30 degrees. For one degree, the value would be $\frac{30\ 0}{30}$, or 1 0 Since there is a difference of 10 degrees between 60 degrees and the temperature of the feed water, we must add to q for 60

degrees 1 0×10, or 10 0, making 28 1+10 0, or 38 1. This is the required q for 70 degrees, or the number of B T.U. in one pound of the feed water as it enters the boiler. Next, obtain the number of B.T.U. in one pound of the water in the boiler after being raised to 140 pounds absolute pressure. In Table III the q corresponding to 140 pounds absolute pressure is found to be 324 4 B T.U. Since the feed water contained 38 1 B.T.U., the number of B.T.U. that has been added in raising one pound of the water from 70 degrees to 140 pounds pressure is 324 4−38 1, or 286 3 B.T.U., the required result.

Since the steam formed contains some moisture, its quality being 99 per cent, it follows that Q, the number of B.T.U. required to vaporize one pound of the water under the given conditions, would be $q+xr$. In the first part of the problem q was found to equal 286 3 B T.U. The value of r for 140 pounds absolute, as obtained from Table III, is 867 4.

$$\begin{aligned} Q &= 286\ 3+(\ 99\times 867\ 4) \\ &= 286\ 3+858\ 73 \\ &= 1145\ 03 \text{ B.T.U.} \end{aligned}$$

This result represents the total heat units necessary to make one pound of steam under the given conditions.

Feed Water Temperature. The above problem brings out a point that has not been noted, viz, the method to follow when the temperature of the feed water is other than 32 degrees, q in the equation $Q=q+x\,r$ being the heat of the liquid corresponding to the pressure, taking 32 degrees as standard. When the feed water at the outset is of a higher temperature than 32 degrees, it is obvious that on account of this higher temperature the number of heat units required to raise it to the boiling point will be less. It follows, therefore, that the above expression should be modified in order to apply to feed water of any temperature. That is, if t is equal to the temperature of the feed water from which the steam is to be made, and q_1 equals the corresponding heat of the liquid, the expression for Q may be modified so as to read

$$Q=q+x\,r-q_1$$

or, taking $q+x\,r=H$, the total amount of heat units added is

$$Q=H-q_1$$

Likewise, the formula for superheated steam may be changed to the form

$$\begin{aligned} Q &= q+r+.48\ (t_s-t)-q_1 \\ &= H+.48\ (t_s-t)-q_1 \end{aligned}$$

Example. How many B.T.U. must be added to one pound of water at 177°F. to transform it into steam at 145.5 pounds gauge pressure and a temperature of 480° F.?

SOLUTION. From inspection, it is evident that the result will be superheated steam, since 480° F. is higher than the temperature corresponding to the 145 5 pounds gauge pressure. This being true, the above formula for superheated steam must be used. Assuming the atmosphere pressure to be 14.5 pounds per square inch, the absolute pressure will be 160 pounds. The value of q corresponding to this pressure is 335 3, r is 858 8, t_s is 480 degrees, t is 363 62, and q_1 is 145 0. Substituting these values in the above formula, we have

$$Q = 335\ 3 + 858\ 8 + 48\ (480 - 363\ 62) - 145\ 0$$
$$= 1194\ 1 + 55\ 86 - 145\ 0 = 1104\ 96 \text{ B.T.U}$$

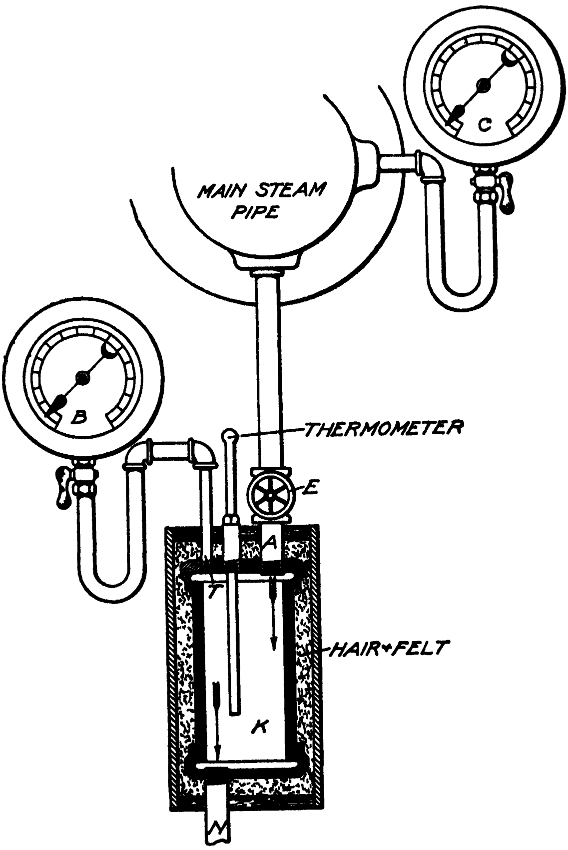


Fig 39 Throttling Calorimeter Connected for a Test

Calorimetric Measurements. In the discussion of the properties of steam and the use of the steam tables, the term *quality of steam* was referred to and was used in every instance where saturated

steam was dealt with. Since it is necessary that the quality of steam be known in all calculations dealing with the amount of heat in a pound of steam, some means must be employed for determining the quality.

Throttling Calorimeter. An apparatus known as a throttling calorimeter was devised in the early eighties by Professor C. H. Peabody for making this determination. It has been widely used since that time and is considered a simple and efficient means for obtaining the required results.

The operation of the throttling calorimeter is based upon the principle that saturated steam will become superheated if the pressure is reduced by throttling without loss of heat. The calorimeter, Fig. 39, performs the above function within certain limits, as will be evident from a description of its action. It consists of a closed metallic cylinder *K*, having a steam inlet *A* and an outlet *N*; a thermometer well, made of suitable material, is provided at *T*. Two gauges *B* and *C* are used, *B* being attached to the calorimeter and *C* to the steam supply line by means of siphons. A valve *E* is placed between the main steam pipe and the calorimeter so as to regulate the amount of flow of steam into the calorimeter. The nipple *A*, connecting the inlet valve *E* with the chamber *K*, is made of special metal, threaded, and having a well-formed orifice, as shown in Fig. 40.

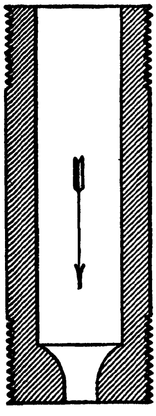


Fig 40 Form of Connecting Nipple

The connection between the steam pipe and the calorimeter should be as short as possible. The cylinder *K* and the connections should be thoroughly covered with asbestos hair felt, or other nonconducting material, in order to reduce the amount of heat radiation. The outlet pipe *N* should be at least 1 inch in diameter for its entire length and it may be larger.

To use the calorimeter, fill the thermometer well *T* with oil or mercury and then insert the thermometer. Attach the gauges and permit steam to enter the cylinder *K* until, say, about 5 pounds pressure is registered on the gauge *B*. This pressure should be kept constant throughout the test by means of the valve *E*. The siphons should be full of water and steam should be permitted to flow through

the apparatus for about ten minutes before taking observations. The observations to be taken are the pressure P of the steam in the main steam pipe, the pressure P_1 of the steam in the calorimeter, the temperature t in the calorimeter, and the barometric pressure P_a. Having this data at hand, the amount of moisture in the steam may be determined by combining the two fundamental equations $Q=q+xr$, corresponding to P, the main steam pipe pressure, and $Q=q_1+r_1+C_p\ (t_s-t_1)$, corresponding to the pressure P_1 in the calorimeter. The absolute pressure in the main steam pipe will be $P+P_a$, and in the calorimeter P_1+P_a. Equating these two expressions, we get

$$q+xr=q_1+r_1+C_p\ (t_s-t)$$

Transposing and dividing through by r, we get

$$x=\frac{q_1+r_1+C_p\ (t_s-t)-q}{r}$$

x being 1 minus the per cent of moisture in the steam, or the quality.

The equation and the method of obtaining the quality of steam will be readily understood by the following example.

EXAMPLE. The pressure P in the main steam pipe equals 69 8 pounds; the pressure P_1 in the calorimeter equals 12 pounds; the pressure P_a of the atmosphere equals 14 8 pounds, the temperature t_s in the calorimeter equals 268.2° F. Determine the quality of the steam.

SOLUTION The absolute pressure in the steam pipe $P+Pa=69\ 8+14\ 8$, or 84 6 pounds The absolute pressure in the calorimeter $P_1+Pa=12+14\ 8$, or 26 8 pounds. The temperature of saturated steam t_1 at 26 8 pounds$=243\ 8$ pounds. It is to be noted that t_1 is the temperature corresponding to the absolute pressure in the calorimeter The total heat $q_1+r_1=1161\ 3$ B T U.; $q=$the heat of the liquid corresponding to $P+Pa=286\ 2$, and r for the same pressure is 896 9

$$\therefore \qquad x=\frac{1161\ 3+0\ 48\ (268\ 2-243\ 8)-286\ 2}{896\ 9}$$
$$=\ 989$$

A throttling calorimeter may be made of pipe fittings, making a simple and convenient apparatus which, if properly constructed and operated, will give good results. Such an apparatus is illustrated in Fig. 41, which also shows the proper method of connection to a steam pipe. Steam is taken from a ½-inch pipe provided with a valve and passes through two ¾-inch tees situated on opposite sides

of a $\frac{3}{4}$-inch flange union, substantially as shown in the accompanying sketch. A thermometer cup, or well, is screwed into each of these tees, and a piece of sheet iron perforated with a $\frac{1}{8}$-inch hole in the center is inserted between the flanges and made tight with rubber or asbestos gaskets, which also act as nonconductors of heat. For convenience a union is placed near the valve as shown, and the exhaust steam may be led away by a short $1\frac{1}{4}$-inch pipe, shown by dotted lines. The thermometer wells are filled with mercury or heavy cylinder oil, and the whole instrument from the steam main to the $1\frac{1}{4}$-inch pipe is well covered with hair felt.

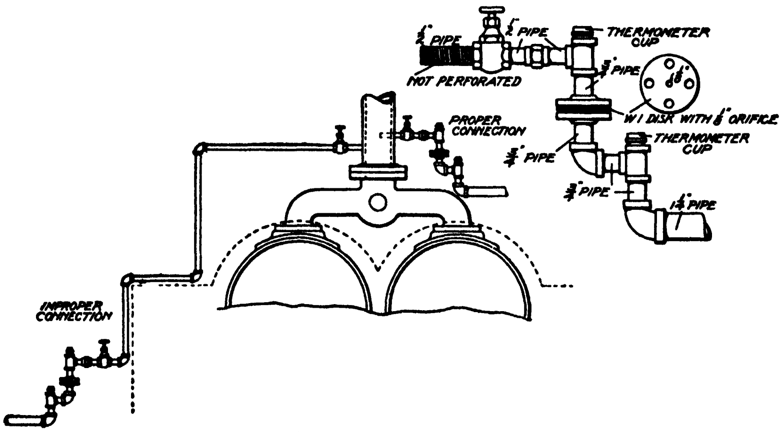


Fig 41 Throttling Calorimeter Made of Pipe Fittings

Great care must be taken that the $\frac{1}{8}$-inch orifice does not become choked with dirt, and that no leaks occur, especially at the sheet iron disk, also that the exhaust pipe does not produce any back pressure below the flange. Place a thermometer in each cup, and opening the $\frac{1}{2}$-inch valve wide, let steam flow through the instrument for ten or fifteen minutes; then take frequent readings on the two thermometers and the boiler gauge, say at intervals of one minute.

Separating Calorimeter. Another type of calorimeter sometimes used in cases where the steam contains from 5 to 10 per cent of moisture, is the separating calorimeter. It works upon the principle that the moisture contained in the steam is liberated by mechanical means. In its usual form, the calorimeter consists of a cylindrical vessel so constructed that the moisture is separated from the

steam and returned, the dry steam passing to a condenser where it is collected and afterward weighed. The separating vessel is provided with a glass gauge and graduated scale which indicates the weight of the moisture taken out of the steam. Having obtained the weight W of the separated water and the weight W_1 of the dry steam, then the percentage of moisture to the total amount of the liquid would be

$$y=\frac{W}{W+W_1}$$

Therefore, the percentage of dry steam would be

$$\begin{aligned} x&=1-y \\ &=1-\frac{W}{W+W_1} \end{aligned}$$

EXAMPLE. Required the weight of steam in the cylinder of a 16×36-inch engine when the piston has moved on its stroke 27 4 per cent of the distance from the h.e. The steam in the cylinder at this instant is under a pressure of 114 pounds gauge, as determined from the indicator card. The piston displacement for the h.e. is 4.188 cubic feet and the clearance on the h.e. is 5.2 per cent. The atmosphere pressure is 14 5 pounds per square inch.

SOLUTION. The first thing required is the volume of steam back of the piston, when the engine has made 27.4 per cent of the stroke. To 27.4 per cent add the clearance 5.2 per cent, making a total of 32.6 per cent, or $\frac{32.6}{100}$ of the whole volume of the cylinder containing steam at the instant under consideration. Since the total piston displacement for the h.e. is 4.188 cubic feet, then the volume of steam to be considered would be $4.188\times\frac{32.6}{100}$, or 1.36 cubic feet. The absolute pressure of the steam in the cylinder would be 114+14.5 pounds, or 128.5 pounds per square inch. Looking in Table III for the weight of 1 cubic foot of steam at 128.5 pounds absolute pressure, we find it to be by interpolation .2867 pounds. Since there are 1 36 cubic feet in the cylinder at 27 4 per cent of the stroke, the weight of steam in the cylinder at the instant in question would be 1 36×.2867, or .3899 pounds.

Volume and Weight of Steam. In considering any problem dealing with the weight of steam in the cylinder or the piston displacement, the per cent of clearance must always be taken into account, as in the problem above.

In studying and analyzing the performance of an engine, it is often desirable to obtain the volume and the weight of steam in the cylinder from the indicator card at the several events, and also to

know the quality of the steam at these points. From the study of the indicator cards and the steam tables, we are now prepared to obtain these several values.

Volume of steam at c.o. in cubic feet is equal to the piston displacement in cubic feet multiplied by the per cent of c.o. plus the per cent of clearance. For example, in the problem given above, the h.e. displacement was 4.188 cubic feet and the h.e. clearance was 5.2 per cent. It is desired to obtain the volume of steam at c.o. which takes place at 34.8 per cent. The sum of the clearance and c.o. per cents is 40. Therefore, the volume of steam at c.o. is 4.188×40 per cent, or 1.675 cubic feet.

Volume of steam at release in cubic feet is the product of the piston displacement and the sum of the per cents of release and clearance.

Volume of steam at compression is found by multiplying the piston displacement by the per cent of compression plus the per cent of clearance.

Weight of steam at c.o. is the product of the weight of 1 cubic foot of steam at the absolute pressure at c.o. and the volume of steam in cubic feet at c.o., both as obtained from the indicator card.

Weight of steam at release is the product of the weight of 1 cubic foot of steam at the absolute pressure at release and the volume of steam in cubic feet at release.

Weight of steam at compression is found in the same manner as that at release.

Re-evaporation or condensation per revolution in pounds is the weight of steam at release minus the weight of steam at c.o. If the answer is positive, it indicates that there is a re-evaporation, and if negative, a condensation.

Re-evaporation or condensation per i.h.p. per hour in pounds is the item just given multiplied by the revolutions per hour and divided by the i.h.p.

Weight of steam per revolution, as determined by weighing, is the total weight of steam used by the engine divided by the total number of revolutions.

Weight of mixture in the cylinder per revolution in pounds is the weight of steam per revolution as determined by weighing plus the weight of steam at compression.

Per cent of mixture accounted for as steam at c.o. is one hundred times the weight of steam at c.o. per revolution, divided by the weight of the mixture in the cylinder per revolution.

Per cent of mixture accounted for as steam at release is one hundred times the weight of steam at release per revolution divided by the weight of the mixture in the cylinder per revolution.

Thermal Efficiency. Having obtained a working knowledge of the properties of steam from the preceding discussion and the problems dealing with the B.T.U. values of steam, we are now ready to consider the thermal efficiency of an engine, but before this can be calculated, several things must be known.

(1) The amount of work done in a unit of time.
(2) The weight of steam used by the engine in the same length of time.
(3) The number of B.T.U. in each pound of steam used.

These quantities must be accurately determined while the engine is in operation.

EXAMPLE. To illustrate what is meant by thermal efficiency, assume an engine which in developing 242 h p uses 13,000 pounds of steam in two hours; steam pressure 186 3 pounds gauge; quality of steam 99 per cent; temperature of feed water 60°F ; and atmosphere pressure 14 7 pounds. Find the thermal efficiency in per cent.

SOLUTION The number of foot pounds of work done in a minute is $242\times33{,}000=7{,}986{,}000$. The number of B T U in one pound of steam at 186 3 pounds gauge, which is 200 pounds absolute, is $q+x\,r-q_1=354\ 3+(843\ 5\times\ 99)-28\ 1$, or 1161 27 B T U. Since the engine is using 13,000 pounds of steam in 2 hours, the amount of steam being used in one minute will be $\frac{13000}{2\times60}$, or 108 333. The corresponding number of B.T U supplied per minute will be 108 333×1161 27, or 125,804 25. Changing this to the equivalent foot pounds of energy by multiplying by 778, we get $125{,}804\ 25\times778$, or 97,875,706 5 as the total energy in foot pounds supplied the engine per minute. Therefore, the thermal efficiency is

$$E=\frac{\text{energy delivered per minute}\times100}{\text{energy supplied per minute}}$$
$$=\frac{7{,}986{,}000\times100}{97{,}923{,}444\ 58}=8\ 15\text{ per cent}$$

The thermal efficiency is expressed by some as the B.T.U. supplied per minute per i.h.p. instead of per cent. Applying this to the problem above, we get for the thermal efficiency

$$E=\frac{13000\times1161.27}{2\times60\times242}$$
$$=519.5\text{ B.T.U.}$$

Generally speaking, the efficiency of a steam engine is spoken of as being so many pounds of steam per i.h.p. per hour. In the problem under consideration, this would give an efficiency

$$E = \frac{13000}{2 \times 242}$$

$$= 26.8 \text{ pounds of steam per i.h.p. per hour}$$

The B.T.U. per i.h.p. per minute varies inversely as the thermal efficiency, so if one value is known, the other can be easily obtained by using the two constants—778 the mechanical equivalent of heat, and 33,000 the number of foot pounds per minute which constitutes a horsepower. If an engine has a thermal efficiency of 100 per cent, it would require $\frac{33000}{778}$, or 42.42 B.T.U. per h.p. per minute. An engine which used 520.1 B.T.U. per minute, as in the above example, has a thermal efficiency of $\frac{42.42}{519.5} \times 100$, or 8.17 per cent.

INTERPRETATION OF INDICATOR CARDS

Theoretical Diagram. As a basis of comparison between indicator diagrams taken from the same engine under different conditions, from different engines, and for design purposes, a theoretical diagram is constructed on the assumption that the expansion curve of a theoretically perfect engine would be that of a hyperbola. Experiments conducted at various times and on a large number of engines substantiate the assumption. The hyperbolic curve has the property that the product of the distances of any point on the curve from the line of zero volume is constant. This when expressed in equation form is

$$P_1 V_1 = C \text{ (constant)}$$

in which P_1 is pressure at c.o. and V_1 is volume at c.o. If P_2 is pressure at release and V_2 is volume at release, then

$$P_2 V_2 = C \text{ (constant)}$$

It follows then that $P_1 V_1 = P_2 V_2$. In this equation, P_1 and P_2 are absolute pressures and V_1 and V_2 include the clearance volume.

To Draw the Theoretical Card. To draw an ideal diagram (see Fig. 42), draw PX equal to the length of stroke and OP equal to the clearance. Draw OY and PA perpendicular to OX and draw YS parallel to OX and at a height corresponding to the boiler pressure.

The line of initial pressure AC is then drawn parallel to YS and is usually taken as from 90 to 95 per cent of the boiler pressure, if there is no special cause for loss. Then take AC as the portion of the stroke at which steam is admitted, so that $\frac{OX}{OR}$ equals the ratio of expansion. The expansion line is considered a hyperbolic curve with OY and OX as asymptotes. To draw the hyperbolic curve.

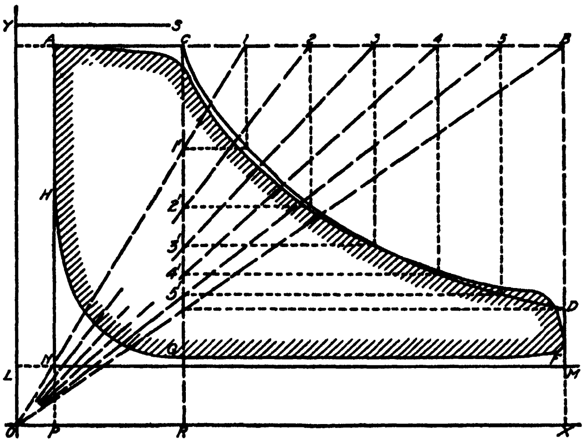


Fig. 42. Ideal Indicator Card

first draw the line ACB parallel to the atmosphere line and FDB and RC perpendicular to it. Then make points *1*, *2*, *3*, *4*, etc., on CB and connect them with the point O. At the points *1'*, *2'*, *3'*, *4'*, etc., where these lines intersect the line RC, draw parallels to CB until they meet perpendiculars from points *1*, *2*, *3*, *4*, etc. The points of intersection of these lines are points on the hyperbolic curve CD, as shown in Fig. 42. Any number of points may be used, but there must be enough to determine the curve. A theoretical compression curve may be drawn in the same manner as an expansion curve, letting the perpendicular to the atmosphere line be drawn from the point of compression instead of from the point of cut-off.

The area $ACDMNH$ is the theoretical card, with a given

boiler pressure and an assumed drop and ratio of expansion. The actual card for the same data would probably appear more nearly like the shaded area which lies mostly within the outline of the theoretical card. In designing engines, it is well to know the ratio of the actual to the ideal card for all types of engines. This ratio varies between .5 and .9 according to the speed, type of engine, and kind of valves.

It will be observed that the actual expansion curve does not coincide with the theoretical curve in Fig. 42. It is a well-known fact that, in the cylinder of a steam engine, the temperature of the steam changes during the stroke. Usually, the piston and valves leak steam more or less; initial condensation takes place at the beginning of the stroke and re-evaporation at the end of the stroke. These factors cause a variation from the true theoretical curve. The object, therefore, in constructing the theoretical diagram is to ascertain where and to what extent these variations occur and to study the causes of the irregularities to the end that the necessary adjustments may be made to eliminate the errors in so far as possible.

In the construction of the theoretical indicator diagram, it is assumed that no loss of heat occurs in the cylinder. It is a well-known fact that as the steam enters the cylinder, some is condensed on account of the comparatively cool cylinder walls. Toward the end of the stroke, the cylinder walls give off heat with the result that either all or a part of the condensed steam is re-evaporated. Hence, the expansion curve of the theoretical diagram would naturally fall below that of the actual curve near the end of the stroke. Speaking in general, a close approximation of the two curves is an indication of good valve adjustment and economical steam distribution. It is, therefore, advantageous to draw the theoretical diagram in order to have something upon which to base an opinion as to the condition of the engine. It would be well when not satisfied with the performance of an engine to construct theoretical indicator cards and compare them with actual cards.

Steam Cards Showing Miscellaneous Troubles. From our study of the indicator diagram, it is evident that a great deal of useful information may be obtained by the correct interpretation of them. Fundamentally, the diagram is to register pressures for given piston positions, so all the information that is obtained in

addition to this, comes from a source of reasoning. A few cards, illustrating information which may be obtained, are given in Figs. 43 to 64, inclusive.

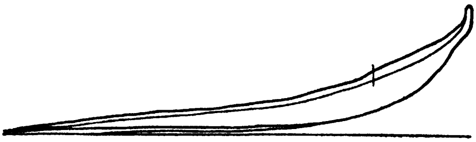


Fig 43 Diagram Showing Improper Valve Lubrication

Valve Trouble. Figs. 43 and 44 illustrate cards taken from the h.e. of one of the cylinders of a locomotive running at thirty miles per hour, using 240 pounds steam pressure, with the reverse lever placed in the second notch ahead of the center position. This locomotive has a superheater, hence, with the high boiler pressure and superheat, trouble was experienced with the lubrication of the valves. This is indicated by the reduced area and distorted card shown in Fig. 43 as compared with that illustrated in Fig. 44. The card shown in Fig. 44 was obtained about twenty minutes later than that illustrated in Fig. 43. In Fig. 44 lubricating oil had been forced into the steam chest. The effect on the card shown in Fig. 43 was caused by the valve clinging to its seat, resulting in a shorter travel and poor steam distribution.

Fig 44 Improvement in Diagram by Use of Lubricant

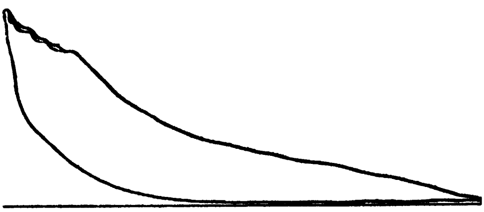


Fig 45 Diagram Showing Sticky Indicator Piston

Sticky Indicator Piston. Figs. 45 and 46 show cards obtained from the c.e. of one of the cylinders of the same locomotive. The wavy appearance of the steam line in Fig. 45 is due to a dry, sticky

Fig. 46. Diagram After Trouble of Fig. 45 has been Removed

indicator piston. Fig. 46 illustrates the appearance of the steam line after the indicator piston had been removed, well oiled, and replaced.

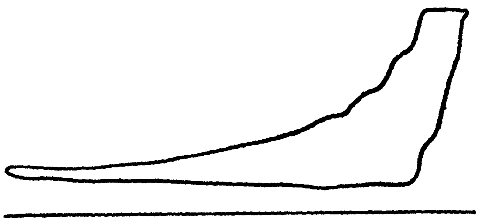


Fig 47. Distorted Card Due to Binding Indicator Piston

Tight Indicator Piston. The cards exhibited in Figs. 47, 48, and 49 illustrate the distortion of the card which may occur when the indicator piston does not fit properly and binds, due to the indicator

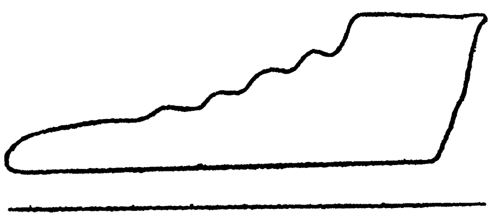


Fig. 48. Distorted Card Due to Binding Indicator Piston

parts not being put together properly. The indicator by means of which these diagrams were obtained had the screw in the bottom of the piston run up so far that the piston rod did not fit down over the projection on the piston, hence perfect alignment was not obtained.

Of the three cases, Fig. 49 is the worst. The area of the card is very much decreased and the back-pressure line is high.

Lost Motion. The effect of lost motion in the connections of an indicator is apparent in the cards illustrated in Figs. 50 and 51. It

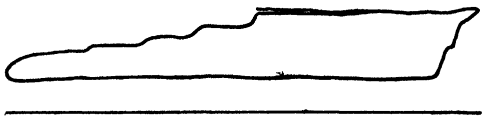


Fig 49 Bad Case of Binding Indicator Piston

is, perhaps, most noticeable in the wave in the expansion line and the height of the back-pressure line.

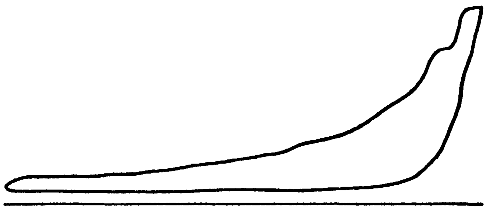


Fig 50 Effect on Diagram of Lost Motion in Indicator Connections

Variable Cut-Off. In Fig. 52 is shown a card taken from a Buckeye engine at a speed below that at which the governor sets. With the engine working under this condition, the greatest c.o. is obtained.

Fig 51. Diagram Showing Effect of Lost Motion in Indicator Connections

Fig. 53 illustrates a card taken from the same engine, running at 200 r.p.m. and with a load slightly under full load. Fig. 54 illustrates another card taken from the same engine operating under a very light load. The c.o. occurs very early. The small area of the card suggests the small amount of work being done in the cylinder.

Long Indicator Cord. Fig. 55 illustrates the effect of too long an indicator cord. Comparing this diagram with those shown in Figs. 52, 53, and 54 taken from the same engine but from the other end

Fig. 52 Card from Buckeye Engine at Low Speed

of the cylinder, the distortion becomes very apparent. Fig. 56 illustrates a distorted card from the same engine, its distortion being due to the cord slipping off from the sector of the reducing motion.

Fig 53 Card from Buckeye Engine at Nearly Full Load

It is to be noted that there is a small loop in the top of this card which indicates too much compression. Oftentimes this loop appears when there is nothing wrong with the indicator or its attachments, but is an indication of disarrangement of the valves of the engine.

Fig 54 Card from Buckeye Engine for Very Light Load

Speed Governing. There are two ways of governing the speed of an engine, namely, by throttling the steam or by varying the point of c.o. to suit the load conditions. The effect of these two methods on the indicator diagram is shown in Figs. 57 and 58. The diagram, Fig. 57, was taken when the speed was maintained constant by changing the point of c.o., this being decreased as the load decreased,

thus reducing the power in the cylinder. The card in Fig. 58 was obtained when the speed of the engine was maintained constant by throttling the steam supply rather than by changing the point of c.o.

Fig. 55. Card Showing Effect of Long Indicator Cord

It should be noted that the area of the card is reduced in the same manner as when the point of c.o. was changed but that the events of the stroke remain unchanged under all conditions of throttling. This

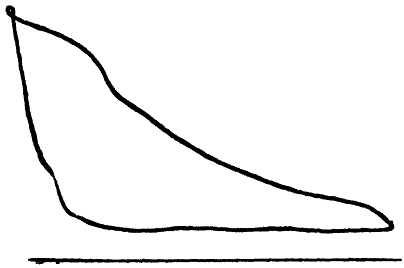


Fig 56. Card Showing Slipping Indicator Cord

is not true, however, of the cut-off governor, because in this type, by changing the point of c.o., the other events of the stroke are affected in some degree.

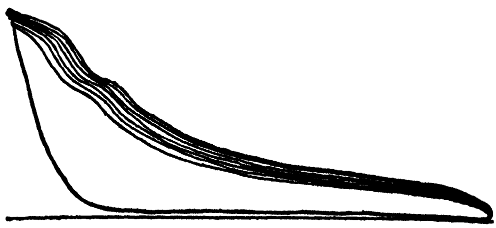


Fig. 57. Card Showing Effect of Changing Cut-Off

Faulty Valve Arrangement. A typical card taken from a Straight Line engine, running at 270 r.p.m. at full load, is shown in Fig. 59.

Aside from illustrating the form of card that this particular type of engine gives, it is of interest because it indicates a faulty valve arrangement. Referring to the figure, it will be seen that admission occurs

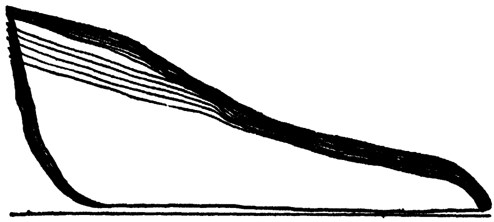


Fig. 58 Effect on Card by Throttling the Engine

at the end of the stroke as indicated at *a*. Late admission is indicated by the sloping admission line, giving the space *b* between the

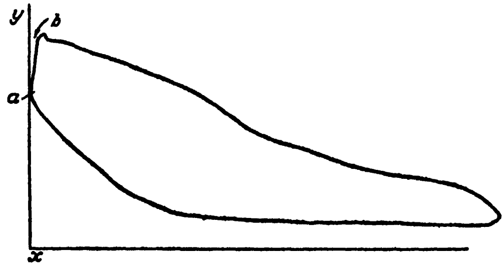


Fig 59. Effect of Faulty Valve Arrangement

end of the stroke and the point where full admission occurs. Early admission would be indicated in the same way, the exception being

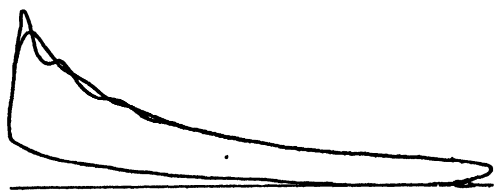


Fig. 60. Card of Gas Engine Operating Under Full Load

that the admission line would slope towards the line *x y* drawn at the end of the card, instead of away from it, as it does in the case illustrated. When retarded admission occurs in a very large degree, the curvature of the admission line is more pronounced.

Gas Engine Cards. It seems desirable to show a few typical gas engine cards, as every engineer may be called upon to indicate gas engines as well as steam engines. Fig. 60 is a diagram obtained from a gas engine operating under a full load of approximately 18 h.p. A 240-pound spring was used in taking the card. Fig. 61 is a diagram taken from the same engine, but operating under different conditions. It shows the change in the diagram produced by throttling the mixture for various loads. This card also shows that the indicator cord stretched slightly, otherwise the different maximum compression points *a* would have fallen on a line perpendicular to the atmosphere line.

In most of the diagrams presented thus far, the errors pointed out were those due chiefly to defects in the operation and attachment of the indicator, or in lubrication.

Fig. 61 Change in Gas Engine Diagram by Throttling the Mixture for Various Loads

Cards Showing Valve Troubles. It is now desired to direct attention directly to defects in valve-setting as shown by the indicator diagram, to the end that suggestions may be given as to how to properly adjust the valves of an engine by the use of an indicator.

The most common faults in the distribution of steam in an engine cylinder can be divided into four classes, viz, admission too early or too late; cut-off too early or too late; release too early or too late; and compression too early or too late.

Late Admission. The diagram, Fig. 59, shows too late admission, as was previously pointed out. If a plain slide valve were used, the reason why admission occurred too late was because the angle of advance was too small. If admission seems too early, the opposite thing is true and the angle of advance should be decreased.

Excessive Back Pressure. The cards shown in Figs. 47, 48, and 49 portray too much back pressure. While in these cards it was due to defects in the indicator, rather than in the engine, yet this excessive back pressure is sometimes found due to inherent defects in the

design of the engine, such as too small exhaust ports or pipes, or to the passage of steam through coils of pipe for heating purposes. Excessive back pressure is an indication of a loss of power and should be kept as small as possible. If the exhaust steam is used for some

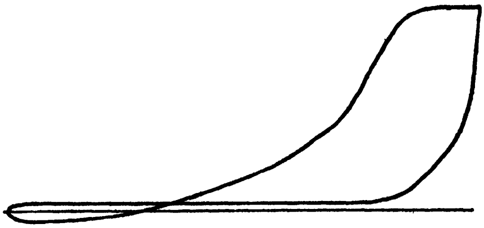


Fig 62. Diagram Showing Effect of Too Early Cut-Off

useful work, such as heating, etc., an increased back pressure above the normal is permissible.

Early Cut-Off. The diagram, Fig. 62, shows the c.o. to come too early. In this case the c.o. is so early that the expansion line extends below the atmosphere line, making a loop. In finding the area of such a card for computing the power, the area of the loop must be subtracted from the total area. In using a planimeter to determine the area, it will automatically make the reduction so the reading will be correct. This loop is frequently spoken of as negative work.

Wire Drawing. Fig. 63 shows a pair of diagrams from a plain slide-valve engine. The admission lines are good. The sloping steam lines show wire drawing due to the slow action of the valve

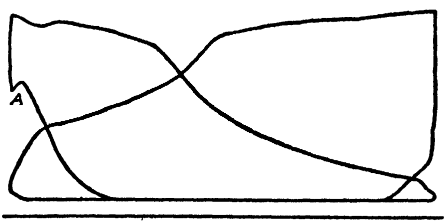


Fig. 63. Pair of Diagrams from Plain Slide-Valve Engine

or too small ports or pipes. This wire drawing decreases the area of the card and indicates a loss. The greatest fault is the inequality of area of the diagram. The late cut-off and consequent late compression of one end causes more area than the too early cut-off

and too early compression of the other end. These cards can be improved upon by adjusting the angle of advance of the eccentric and the length of the valve rod. If the left card were a normal one the hook at *A* might indicate an open cylinder cock.

Early Compression. The diagram of Fig. 64 indicates too early compression. The compression curve extends above the initial pressure line. The area of the loop must be subtracted from the card area when computing the i.h.p. If the cut-off is kept the same and the compression made what it should be, the gain in area would be the area included between the full line and the dotted line plus the area of the loop. The remedy for this case is to decrease the inside lap, which would permit exhaust to occur earlier and compression later.

The amount of compression an engine should have varies with the speed and type. Slow speed engines require less compression or

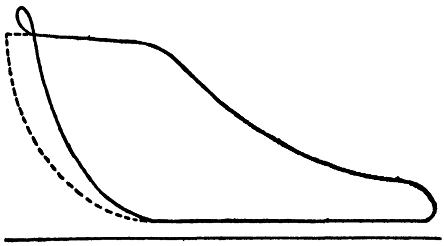


Fig. 64. Diagram Showing Early Compression

cushioning than high speed engines. The exhaust steam should never be compressed higher than the boiler pressure.

If the valve travel is increased, compression is retarded—that is, decreased—and release occurs sooner.

TESTING STEAM ENGINES

In the beginning of this study, it was stated that the indicator had been largely responsible for the refinement of the modern steam engine. In what way the indicator has influenced the development will be evident from the suggestions which follow and from the work involved in the testing of engines. The testing of steam engines requires considerable preliminary work and very careful attention to details. The tests may be made to ascertain whether the valves

are properly set; to determine i.h.p., b.h.p., and f.h.p.; to determine the amount of steam used per i.h.p. per hour, or the commercial efficiency; and to investigate the transference of heat between the steam and the cylinder walls, and losses due to this transference. It should be borne in mind that most of the results sought for are closely allied, so that one complete test may give data sufficient to obtain the value of all the factors mentioned. For instance, if one is seeking the loss due to friction, he must obtain the b.h.p. and i.h.p. and, having these, it is an easy matter to obtain the mechanical efficiency.

Factors Considered. Usually the principal object in testing a steam engine is to determine the cost of power or the effect of such conditions as superheating, jacketing, varying the point of cut-off, varying the point of compression, clearance, steam pressure, etc., upon the steam economy of the engine. We must determine, therefore, first, the cost of fuel, and second, the actual amount of heat used. In either case, the horsepower of the engine must be determined.

The indicated power is determined by means of the indicator, and the actual power delivered, by means of a dynamometer or friction brake. To determine the cost of power in terms of coal, it is necessary to conduct a careful boiler test, usually of twenty-four hours duration.

When the cost is expressed in terms of steam per horsepower per hour, we may follow either of two methods, viz, we may condense and weigh the exhaust steam, or we may weigh the feed water supplied to the boiler. When the object of the test is primarily for an investigation of the performance of the engine, it is best to weigh the condensed steam. This is the method used in the test described herein. An hour under favorable conditions is usually sufficient for such tests. Steam used for any purpose other than running the engine must be determined separately and allowed for.

Probably the most accurate terms in which to state the performance of an engine is in B.T.U. per horsepower per minute. When the cost is expressed thus, it is necessary to measure the steam pressure, amount of moisture in the steam, and temperature of condensed steam when it leaves the condenser. Jacket steam must be accounted for separately. Engines with their boilers, etc., for large plants, are usually built under contract to give a certain efficiency, and their

fulfillment of this contract can be determined only by a complete test of the entire plant. Before beginning the test, the engine should be run for a sufficient length of time in order to limber it up and get it thoroughly warmed. It is of the utmost importance that all conditions of the test should remain constant, especially the boiler pressure and the load. All instruments used in the test should be calibrated before being used, in order to determine the effect of any errors to which they may be subject.

Thermometers. All important temperatures, such as feed water, injection water, condensed steam, etc., must be taken by accurate thermometers, the errors of which have been previously determined and allowed for. Good thermometers sold by reliable dealers are usually satisfactory. Cheap thermometers are of little value in an engine test.

Indicators. The most important and in many respects the least satisfactory instrument used in the test is the indicator. It is subject to an error of 2 to 3 per cent, depending on the conditions. It does not work satisfactorily at more than 400 revolutions per minute. If the indicator is carefully tested under conditions similar to those under which it is used, the errors may be reduced to a minimum, but there will always be some uncertainty. The principal errors to which the indicator is subject have already been mentioned.

Scales. Weighing should be done on standard platform scales. The water may be weighed in barrels provided with large quick-acting drain valves which will allow the water to run out quickly. It is seldom possible to drain barrels completely, and so it is best to let out what will run freely, then shut the valve and weigh the barrel. This we call "empty" weight, and deducted from the weight "full" evidently gives us the true weight of water.

If not convenient to weigh the water, it may be measured in tanks or receptacles of known capacity, and the temperature taken, allowing the proper weight per cubic foot for water at that temperature; or it may be determined by meters.

Meters. Water meters are of two kinds, viz, those that record the amount of water by displacement of a piston, and those in which the flow is recorded by means of a rotating disk. Piston water meters can be made very accurate, and if working under fair conditions of service, they may be relied upon to a close degree. The

chief error in a meter arises from the air that may be in the water. To reduce this error to a minimum, the meter should be vented so as to allow the air to escape without passing through the meter. Rotary meters are good enough for very rough work, but are seldom sufficiently accurate for a careful engine test. So far as possible weirs should not be used in engine work. They may be fairly accurate under certain conditions, but a very little oil in the water may affect them seriously. They may sometimes be used to measure the discharge from a jet condenser, for then the volume is so large that the actual error is proportionately small. The use of meters for testing purposes should always be discouraged. When used, however, they should always be carefully calibrated under as nearly as possible the same conditions as existed during the test.

Gauges. Pressures should be measured on good gauges that have been recently tested. The atmospheric pressure should be read from the barometer, and for accurate work this pressure should be used. For ordinary work, 30 inches of mercury, or 14.7 pounds, may be used.

Calorimeters. When using superheated steam, it is sufficient to take the temperature and pressure in the steam pipe, but if saturated steam is used, we must determine the amount of moisture it contains. This is done by means of a calorimeter such as has previously been described.

Prony Brakes. Any of the forms of friction brake described will answer the purpose. For smooth and continuous running, it is essential that the brake and its band be cooled by means of water and that some lubrication be applied to the surface of the brake wheel. The water may either circulate in the rim of the wheel or around the brake band, but it must not come in contact with the rubbing surfaces.

Original Type. The most common form of brake used is some modification of the Prony brake as illustrated in Fig. 65. This is one of the simplest forms of absorption dynamometer. The two wood blocks A and C are held together against the rim of the pulley P by bolts. The thumb nuts, E E, are used to adjust the pressure. By means of the bolts, the arm L is held to the upper block. From this arm is suspended the ball weight B which, by sliding along the arm, counterbalances the weight of the arm and pan at the other end. The pulley revolves at the required speed in

the direction indicated by the arrow. The bolts are tightened until the lever remains stationary in a horizontal position when a known weight W is hung at the end. Suitable stops must be arranged at

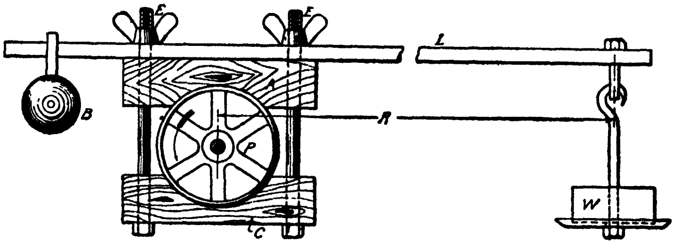


Fig 65. Original Form of Prony Brake

the outer end of lever L to prevent an accident in case the brake should happen to grip the wheel and cause the weight W to be thrown over.

The amount of work absorbed by the brake depends upon the weight W, the length R, and the speed. It is independent of the diameter of the pulley and the pressure of the blocks because the moments of forces about the center of the pulley are equal when the

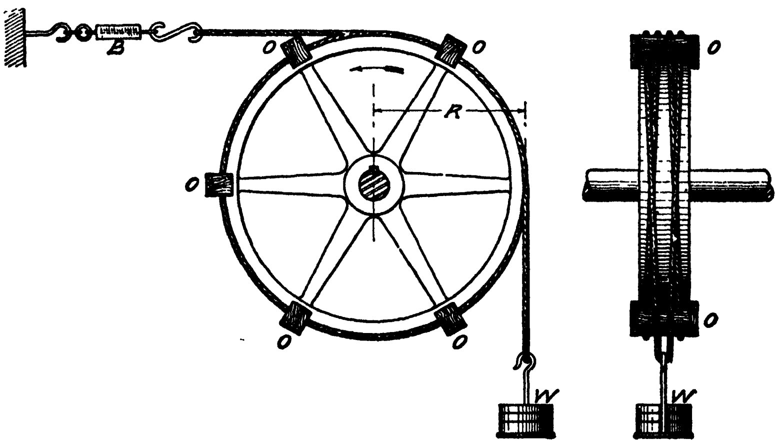


Fig. 66. Rope Form of Prony Brake

lever L is horizontal. Letting f equal the coefficient of friction, p the pressure of the blocks, and r the radius of the pulley, we have

$$f\,p\,r = W\,R$$

The work done at the face of the pulley equals the tangential force between the block and the wheel multiplied by the distance passed over, which also equals weight W multiplied by the number of feet W would move through if it were free to rotate.

Let N be the number of revolutions per minute. Then the distance passed through per minute equals $2\pi rN$ and the work done equals $2\pi rNfp$. Then as $fpr = WR$, the work done at the rim of the pulley equals the left-hand side of the equation multiplied by $2\pi N$, and to keep both sides equal we multiply WR by $2\pi N$. Then the work done per minute is obtained from the expression $2\pi NWR$.

$$\text{b.h.p.} = \frac{2\pi NWR}{33000}$$
$$= .0001904\ NWR$$

EXAMPLE. A Prony brake having an arm 4 feet long attached to the pulley of an engine sustains a weight in the scale pan of 50 pounds when the speed of the engine is 300 r p.m. Find the brake horsepower.

$$\text{b.h.p.} = .0001904 \times 300 \times 50 \times 4$$
$$= 11\ 424$$

Rope Type. The rope brake shown in Fig. 66 is easily constructed of material at hand and being self-adjusting needs no accurate fitting. For large powers, the number of ropes may be increased. It is considered a most convenient and reliable brake. In Fig. 66 the spring balance B is shown in a horizontal position. This is not at all necessary; if convenient the vertical position may be used. The ropes are held to the pulley or flywheel face by blocks of wood O. The weights at W may be replaced by a spring balance if desirable.

To calculate the brake horsepower, subtract the pull registered by the spring balance B from the load at W. The lever arm R is the radius of the pulley plus one-half the diameter of the rope. The formula for power absorbed is

$$\text{b.h.p.} = \frac{2\pi RN(W-B)}{33000}$$

$$= .0001904\ R\ N\ (W-B)$$

If B is greater than W, the engine is running in the opposite direction. In this case the formula becomes

$$\text{b.h.p.} = .0001904\ R\ N\ (B-W)$$

EXAMPLE. A rope brake is attached to a gas engine brake wheel. The average reading of the spring balance is 8 pounds when W is 80 pounds. If the radius of the brake wheel is 28 inches and the rope 1 inch in diameter, what is the b.h p. when the engine makes 350 revolutions per minute?

$$R = 28 + \tfrac{1}{2} = 28\tfrac{1}{2} \text{ inches} = \frac{28.5}{12} \text{ feet}$$

Then from the equation for brake horsepower, we have

$$\begin{aligned} \text{b.h.p} &= .0001904\ R\ N\ (W-B) \\ &= 0001904 \times \frac{28\ 5}{12} \times 72 \times 350 \\ &= 11.4 \end{aligned}$$

If both the indicated horsepower and the brake horsepower are known, the power lost in friction may be found by subtracting the b.h.p. from the i.h.p.

Modern Band Type. The two forms of brakes shown in Figs. 65 and 66 serve their purpose very well but are not very durable.

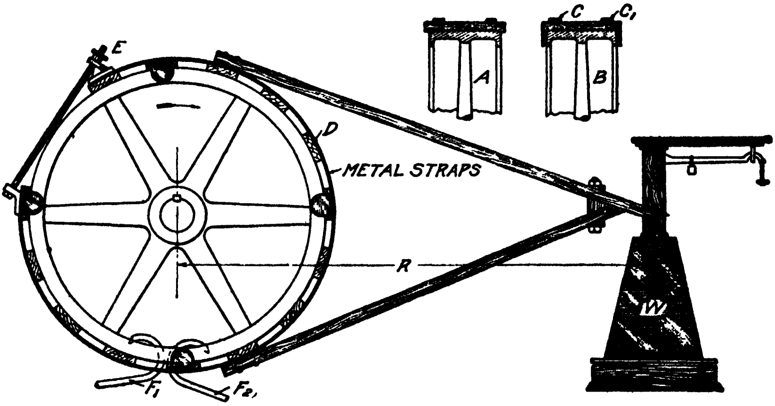


Fig. 67 Modern Band Form of Prony Brake

When it is desired to make repeated tests of an engine for a considerable period of time, or when it is desired to keep the machine in readiness for tests at all times, as in experimental engineering laboratories, it is better to provide a brake of the form illustrated in Fig.

67. This brake is made up of two metal straps CC_1, as shown in the cross-sectional view. Attached to these metal straps are a number of wood blocks placed at regular intervals. These blocks are made of hardwood and form the rubbing medium of the brake. The brake band and blocks are held in place on the pulley by having metal clips extending down the side of the pulley for a fractional part of an inch. The brake is tightened up by means of the hand wheel *E*. The pipe F_1 delivers water to the rim of the wheel for keeping it cool. Pipe F_2 is arranged to scoop up the water from the rim, thus keeping the rim of the wheel filled with cool water. Ordinarily pipe F_2 is not needed. The water in the rim will never be heated above the boiling point and this temperature will do no harm. When the engine is running in the direction indicated by

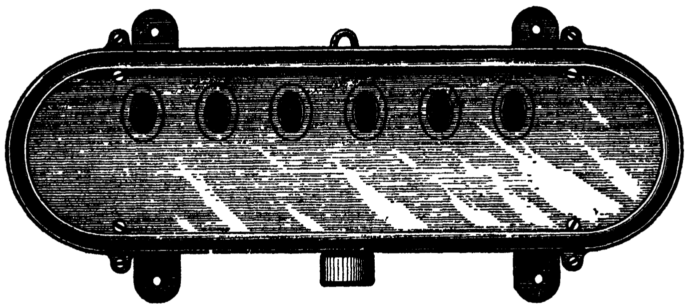


Fig 68 Revolution Counter and Recorder

the arrow, the tendency is for the brake band to move in the same direction, but the V-shaped arms resting upon the platform scales prevent this, and the amount of pressure *W* exerted by the brake lever is weighed by the scales. Hence, one can at any time easily determine the work delivered to the brake. This form of brake is shown in application on a Buckeye engine in Fig. 10; the scales are not used, but instead the brake arm is connected to a chain, which runs over a quadrant to which it is attached. Attached to this quadrant is an arm that carries a weight *B* and a pointer *E*. The pointer indicates the pounds pull on the graduated arc *C*. By careful calibration, the arc *C* is graduated in pounds. In the brake shown in Fig. 67, the pressure *W* on the scale must be corrected before using the brake horsepower formula, for the unbalanced weight of the brake arm. If the brake band is supported on a knife edge imme-

diately above the center of the engine shaft, and the outer end of the shaft then weighs W_1 pounds, the brake horsepower formula would be

$$\text{b.h.p.} = .0001904\, RN\,(W - W_1)$$

Speed Counter. In finding the b.h.p. or i.h.p. of an engine, it is necessary to know the number of revolutions the engine makes in a minute. This speed is usually designated as r.p.m. In order to obtain the correct r.p.m., an instrument known as a revolution counter is usually attached to some revolving or reciprocating part of the engine. A common form of such a counter is shown in Fig. 68.

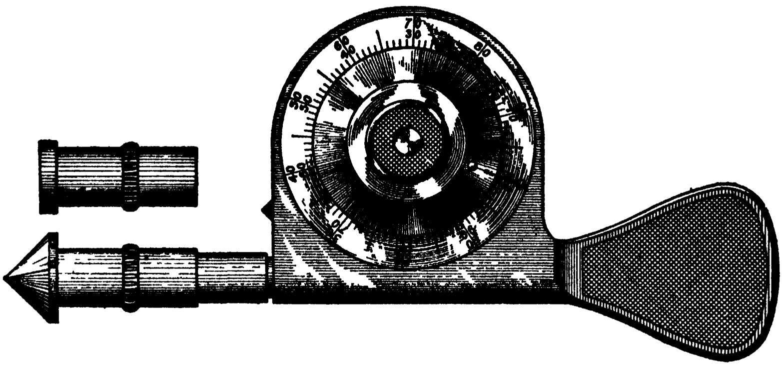


Fig 69 Standard Form of Speed Counter

The actuating motion of the engine or other machine to which the counter is to be attached, is generally communicated by a rod or bar moving in the same general direction of its length, and the lever should be connected to such rod at a right angle when such rod is in the middle of its movement. It should not be clamped rigidly to the shaft until the latter is turned so as to bring the pawl to the middle of the stroke. It may be determined, practically, by opening the lid of the counter and watching the movement of the pawl as the shaft is rotated, when the middle point of its travel can be easily fixed. When in this position, clamp the crank to the shaft by means of the set screw.

This arrangement provides for the utilization of the entire motion of the actuating rod at the angle of greatest effectiveness in moving the mechanism of the counter; and if for any reason the

movement of the rod is shorter than its longest possible stroke—as might happen in the case of a direct acting pump—there would still be ample motion to insure a correct count.

This counter is adapted to either right or left rotary or reciprocating motions and is capable of 500 revolutions per minute with safety to the machine and accuracy in the enumeration.

The shaft through which the actuating force is applied may extend from the counter either on the right-hand or left-hand side, as desired.

A very simple form of speed counter is illustrated in Fig. 69. It has a rubber tip which is held in the center of the engine shaft. The motion of the engine shaft is transmitted to the shaft of the counter which drives through a system of gears a pointer, the latter indicating on a graduated dial the number of revolutions made in a given time. When well made, this is a very accurate instrument and may be read with reliability for speeds up to as high as two or three thousand r.p.m.

INDICATOR TROUBLES AND REMEDIES

Necessity for Care in Using Indicator. The steam and gas engine indicator is an extremely valuable instrument for engineering purposes when used intelligently, but when in the hands of a careless inexperienced operator the results obtained may be little short of worthless. The instruments constructed by reputable manufacturers are reliable for the purposes for which they were intended and are indispensable in a steam or gas engine power plant of any considerable size. For scientific and investigative purposes the most reliable instrument should be used and the operator should be careful and experienced, in order that the best possible results may be obtained. Many operators make use of the indicator with a desire to secure reliable information and in many instances are sincere and painstaking in their efforts, but, unless they give proper attention to certain fundamental precautionary rules, the accuracy of the results secured may be questioned.

Attachment of Indicator. *Short Connections Desirable.* As has been previously stated, in order to secure reliable diagrams, the indicator should be attached as close to the cylinder as conditions of the particular installation will permit. Long pipe connec-

tions result in unreliable indications. Generally speaking, other conditions remaining the same, the shorter the connections the more accurate the results. Most modern steam engines are now made with suitable holes which are tapped for indicator connections. When the cylinders are not drilled and properly tapped for receiving the indicator, the engineer in charge should be competent to do it under the directions here given.

Conditions Affecting Location of Indicator. Before deciding just where the holes should be drilled, it is desirable that all conditions of the case be carefully studied with a view of devising the whole plan for indicating the engine. It usually happens that the reducing motion, or drum motion as it is sometimes called, can be erected more advantageously in one position relative to the engine than another, or one kind may be better adapted for a given place than another. The type of engine, location of the steam chest or valves, the kind of cross-head and the best means of attaching to it, and the position of the eccentric, its rods, and connections, all should be given careful consideration in determining the best places to locate the indicator. A free passage for steam to the indicator is of prime necessity and a location of the indicator insuring convenience in operation is desirable. The instrument can be used in a horizontal position but in taking diagrams it is more convenient when in a vertical position. Then again, the vertical position is that in which it would most probably be calibrated and for this reason alone is preferable. A prominent manufacturer gives the following directions for drilling cylinders to receive indicators:

Mounting Indicator on Cylinder. When drilling holes in the cylinder the heads should be removed if convenient, so that one may know the exact position and the size of the ports and passages and be able to remove every chip or particle of grit which might otherwise do harm in the cylinder or be carried into the indicator and injure it. When the heads cannot be taken off, it can be arranged so that a little steam may be let into the cylinder when the drill has nearly penetrated its shell, so that the chips may be blown outward—care being taken not to scald the operator.

It is essential that the holes be drilled into the clearance space at points beyond the travel of the piston so as not in any way to obstruct the passage of steam to the indicator. The most common practice in the case of horizontal engines is to drill and tap the holes in the side of the cylinder at each end. On certain types of horizontal engines, it is possible to drill and tap into the top of the cylinder at each end, in which case the indicator cocks can

be screwed directly into the holes. On vertical engines, the upper indicator is frequently connected into the cylinder head, although better results will be obtained if both holes are drilled and tapped in the side of the cylinder.

Reducing Motions. It sometimes happens that in an effort to get quick results a makeshift type of reducing motion, or drum motion, is resorted to, with the almost inevitable result that the diagrams secured by its use are extremely faulty and in some cases worthless. In the long run, the most satisfactory results are secured if some form of approved reducing motion, such as has already been described, is used. Results of experience have shown that diagrams varying in lengths from $2\frac{1}{2}$ to $3\frac{1}{2}$ inches, depending upon the speed of the engine, are most satisfactory. These lengths have been found long enough to admit of all useful divisions, and the movement of the card is slower and the tracing correspondingly more delicate and accurate than if a longer card is made. These facts should be borne in mind in designing and proportioning the reducing motion.

Drum Spring Tension. A great many operators give no attention to the tension of the drum spring, using the same adjustment for testing engines operating at wide ranges of speeds. Theoretically speaking, there is only one correct drum spring tension for one speed, other conditions remaining unchanged, but the refinement need not be carried to this point. However, it is a matter which should at least receive some attention. For a particular installation and speed the tension should be a sufficient amount, and no more, to overcome the friction of the pencil on the paper and maintain, at all times, the indicator cord taut. Any very great amount of tension, in addition to that necessary, not only affects the wearing qualities of the instrument but shortens the life of the indicator cord.

Adjustment of Guide Pulley. As has been heretofore explained, the object of the guide pulley is to properly conduct the indicator cord from the drum to the reducing motion. It is such an insignificant piece of mechanism that it is frequently overlooked by the inexperienced operator. Whenever this occurs, the diagrams are usually unsatisfactory in many respects, as can readily be seen, and in a very short time there results a broken indicator cord which, under certain conditions, is extremely

difficult to repair. The adjustment of the guide pulley is one of the first adjustments which should be made in setting the indicator for taking cards.

Adjustment of Pencil Pressure. In the early forms of indicators, the diagram was drawn on plain paper by means of a graphite pencil, the pencil being sharpened to a fine round point by means of a knife or fine file. The graphite pencil can still be used but a more satisfactory result is obtained by the use of a brass point for a pencil in connection with chemically prepared paper, known as metallic paper. In either case the result desired is the securing of a light distinct diagram, that is, a diagram which is distinct yet made by using a pencil pressure no greater than is absolutely necessary. This is a matter that is quite generally overlooked by the average operator, the tendency being to obtain a diagram showing much contrast. The operator should always remember that within certain limits the lighter the line the more accurate the results. If too great a pressure is employed between the pencil and paper, the pencil will lag on both ascending and descending pressures, with the result that the diagram will be too small and will not represent the true power of the engine. It will not only give an incorrect indication of the power and pressures but also will improperly represent the true location of the various events of the cycle.

Miscellaneous Precautions. *Importance of Rules for Use of Indicator.* It should be the sincere effort of every operator to secure the very best results possible when making use of the indicator. To this end very careful attention should be given to the brief rules prepared by the manufacturers for assembling and manipulating the instrument. At least attention should be given to these general directions until one becomes thoroughly familiar with the apparatus. For example, in the case of the Crosby indicator, the directions given on pages 30 to 33 of this text, inclusive, should be thoroughly digested and mastered. No matter where the indicator is made or by whom, the operator should adopt a correct and regular method of procedure in its use so that it becomes a habit.

Care in Handling Indicator. Perhaps one of the chief reasons that the indicator receives so many damaging knocks and blows is

the manner in which it is removed from its carrying case and assembled. On opening the carrying case preparatory to taking diagrams, the indicator should at once be lifted out and attached to the indicator cock where it will be securely held while the spring and parts are being connected and adjusted. Before attaching the indicator, however, the cock should be opened for a very brief time and steam be permitted to blow through so as to blow out any foreign matter which might be detrimental to the correct action of the instrument. When the indicator is not in use, it is preferable to place a cap on the indicator cock, which cap is usually furnished with the indicator.

In lifting the indicator from one position to another, never do so by taking hold of the drum as many instruments are rendered useless by carelessness in this regard. In some designs the drum is not held in position by means of a small thumb nut but slips off very easily.

Lubrication. The question of proper lubrication is one which should be handled intelligently. Always before placing the piston and spring in the proper working position, the piston should receive a generous supply of oil. For steam engine work a good grade of valve oil should be employed, while for gas engine work a good quality of gas engine oil should be used; machine oil should never be used. For air compressor work and hydraulic work a high grade of light oil should be used. Occasionally the pencil mechanism should receive oil, which should be light and offer little tendency to cause gumming. Cases are on record in which diagrams were taken with the indicator piston lubricated with the wrong kind of oil and much time was spent and trouble experienced in an effort to diagnose an apparent error which did not exist. Hence the necessity for proper lubrication.

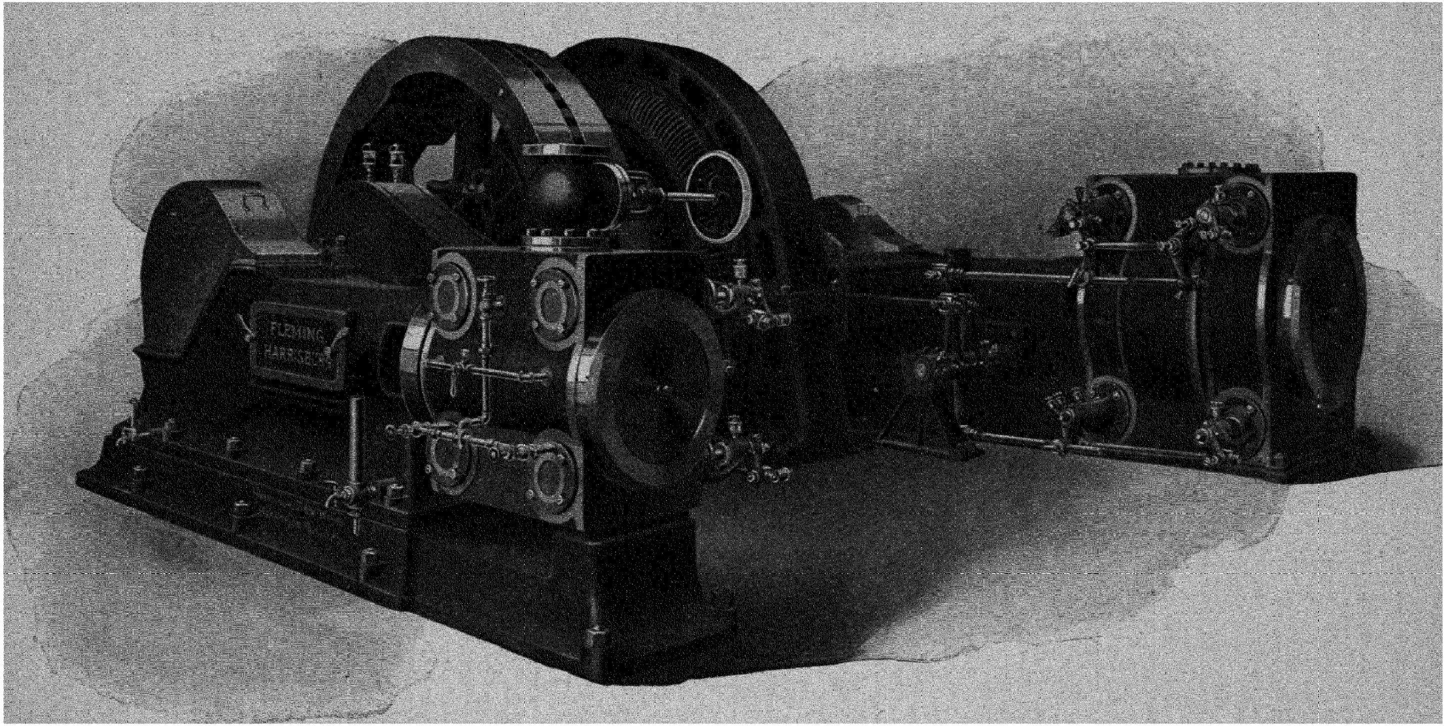
Causes of Incorrect Indication. In taking diagrams many things may happen which will result in incorrect indication. In taking a series of cards, if one notices a card which is much shorter than all others, it may be due to one of two things: either the indicator cord is not properly connected to the reducing motion or by some means it has become too long.

It will sometimes happen that the pressures on one card are much smaller than on others in the series. When this occurs, if

there has been no material change in steam pressure, it is very probably caused by the indicator cock being opened only partially. A leaky indicator cock is always a source of much annoyance. It not only causes incorrect indications of pressures, especially on the exhaust side of the diagram, but produces an irregular atmospheric line which otherwise would be straight.

A diagram which shows an abnormal back pressure, when the engine is known to have but very little back pressure, is most probably due to a loosely connected piston or pencil mechanism, unless it is caused by a sticky piston. In the latter case, however, other indications on the diagram would probably reveal the facts.

Modifications of Indicator for High Speeds. The indicator as usually constructed will give satisfactory results for all ordinary speeds. It cannot be used successfully for speeds above 450 revolutions per minute. For the higher speeds it is necessary to use a heavier spring than would be needed for the same pressure at lower speeds. For such high speeds it is also desirable to use a reducing motion proportioned so as to give a card having a length not to exceed 2 or 2½ inches. Best results are secured when the indicator is used at speeds below 200 revolutions per minute.



FLEMING-HARRISBURG CROSS-COMPOUND HEAVY-DUTY CORLISS VALVE ENGINE
Courtesy of Harrisburg Foundry and Machine Company, Harrisburg, Pennsylvania

VALVE GEARS

VALVE CHARACTERISTICS

Function. Steam enters the cylinder of a steam engine through ports which must, in some manner, be opened and closed alternately, in order to admit and exhaust the steam at the proper time. To accomplish this purpose, one or more valves are moved back and forth across the port openings. A complete understanding of the valve and valve gear is essential to the engineer as well as to the designer, for even though a valve be properly designed, the economy of the engine may be seriously impaired by improper valve setting. The design and adjustment of these valves play a very important part in the efficient action of the steam engine.

A valve gear is a mechanism consisting of a combination of slotted links, eccentrics, rods, levers, and other devices, designed to operate valves of various types. The valve gear is separate and distinct from the valve. It operates the valve or valves but, strictly speaking, is not a part of them. This being true, one type of valve gear may be applied or used in connection with several different types of valves. For instance, the Stephenson gear may be used to operate a plain slide valve on one engine, a piston valve having either inside or outside admission on another, while a third may be attached to a more complicated form of valve mechanism. It should be borne in mind, therefore, that the valve gear is a separate and distinct part of the steam engine and that its function is to impart motion to the valve or valves.

The valves, in turn, perform the following functions during the engine cycle:

(1) *Admission.* This begins when the valve opens to admit steam to the cylinder.

(2) *Cut-Off.* This is the point at which the valve closes to cut off the admission of steam.

(3) *Expansion.* This takes place from cut-off to release.

(4) *Release.* This begins when the exhaust port is opened.
(5) *Compression.* This begins when the exhaust port is closed.

There may be a single valve to regulate admission and exhaust or there may be a double set of valves, one set to admit the steam at each end and another to release it. The valve may have a plain reciprocating motion, moved either by a rod or by some device that releases at the proper time, allowing the port to close suddenly under the influence of counterweights, springs, or vacuum dashpots. To the first class belong the plain slide valve and its modification of piston valve, gridiron valve, etc.; to the second class belong such valves as the Corliss, the Brown, and others.

The simplest type of valve is the plain slide, or *D*, valve as shown in Fig. 1, in which *V* is the valve, *R* is the valve rod, *K* is the exhaust cavity, P_1 and P_2 are the steam ports, *E* is the exhaust port, *A B* is the valve seat, and *D M* are the bridges of the valve seat. The valve seat must be planed perfectly smooth, so that steam pressure on the valve will make a steam-tight fit and cause as little friction as possible when the valve moves back and forth. Furthermore, the length of the seat *A B* must be a little less than the distance from the extreme right-hand position of the right-hand edge of the valve to the extreme left-hand position of the left-hand edge of the valve. This allows the valve at each stroke to slightly overtravel the seat, thus keeping it always worn perfectly flat and smooth. If the valve seat were not raised slightly above the rest of the casting, or if it were too short, the constant motion of the valve would soon wear a hollow path in the valve seat, and it would cease to be steam tight.

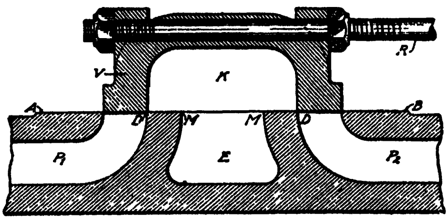


Fig. 1. Plain Slide, or D, valve

Eccentric. The valve usually receives its motion from an eccentric, which is simply a disk keyed to the shaft in such a manner that the center of the disk and the center of the shaft do not coincide. It is evident that as the shaft revolves, the center of this eccentric disk moves in a circle about the shaft as a center, just as if it were at the end of a crank. The action of the eccentric is equiva-

lent to the action of a crank whose length is equal to the distance between the center of the eccentric and that of the shaft.

Fig. 2 represents the essentials of an ordinary eccentric. O_1 is the center of the shaft, O_2 is the center of the eccentric disk E, and S is a collar encircling the eccentric and attached to the valve rod R. As the eccentric turns in the strap, the point O_2 moves in the dotted circle around O_1 and the point A_1 also moves in a circle. When half a revolution is accomplished, the point O_2 will be at O_3, the point A_1 will be at A_2, and the eccentric strap and valve rod will be in the position indicated by the dotted lines. The distance O_1O_2 of the

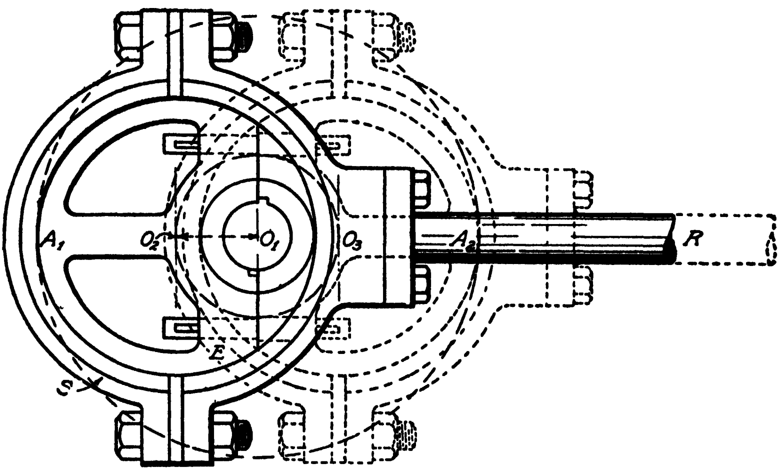


Fig. 2. Details of Ordinary Eccentric

center of the eccentric from the center of the shaft is known as the *eccentricity*, or *throw*, of the eccentric. The *travel* of the valve is twice the eccentricity.

Since the eccentric transmits the motion of the revolving shaft to the valve, it will be necessary to study the relative motions of crank and eccentric in order to obtain a clear idea of the steam distribution. This relation will be developed in connection with the discussion of the valve action which follows.

Valve Motion. *Valve without Lap.* Fig. 3 shows a section through the steam and exhaust ports of an engine, together with a plain slide valve placed in mid-position* and so constructed that

* A valve is in mid-position when the center line of the valve coincides with the center line of the exhaust port.

in this position it just covers the steam ports. Referring to Fig. 1, which shows the same type of valve drawn to a larger scale, suppose the valve is moved a slight distance to the right; the port P_1 is then

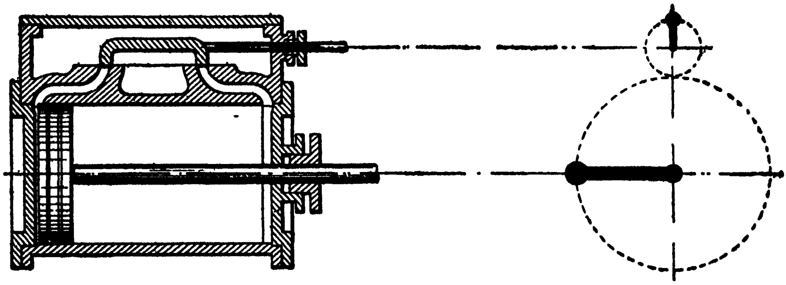


Fig. 3. Cylinder Details_Showing Plain Slide Valve without Lap in Mid-Position

uncovered and opened to the live steam which enters the cylinder and causes the piston to move. Since the two faces of the valve are just sufficient to cover the steam ports, it is evident that as the port P_1 opens to live steam, the port P_2 opens to the exhaust. The ports are closed only when the valve is in mid-position. This allows admission and exhaust to continue during the whole stroke. With such a valve, there is no expansion or compression; the indicator card is a rectangle, and the m. e. p. is equal to the initial steam pressure, assuming no frictional losses in the steam pipe or condensation in the cylinder.

For a theoretical discussion of valve motion, it is assumed that the eccentric rod moves back and forth in a line parallel to the center

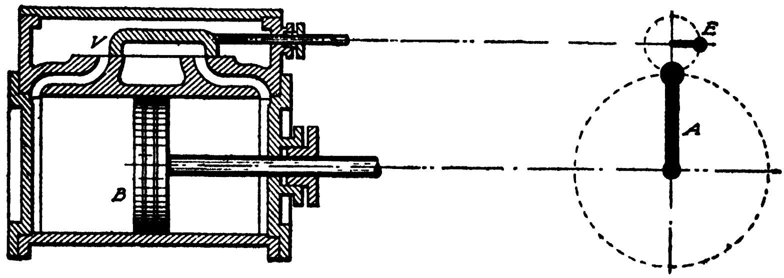


Fig. 4. Position of Piston and Valve in Cylinder Shown in Fig 3, after One-Half Stroke

line of the engine. This is not the case in practice, for the eccentric rod always makes a small angle with the center line, just as the connecting rod does, but the eccentricity is so small in comparison with

the length of the eccentric rod that the angularity of the eccentric rod is very much less than the angularity of the connecting rod, and its influence may be neglected without appreciable error.

When the valve shown in Fig. 3 is in mid-position, the crank is on dead center, the eccentric is set at right angles to it, and the piston is just ready to begin the stroke.

Fig. 4 shows the relative positions of the crank *A*, piston *B*, eccentric *E*, and valve *V*, when the crank has made a quarter turn or the piston has moved to about half-stroke. The eccentric is now in its extreme position to the right, the valve has its maximum dis-

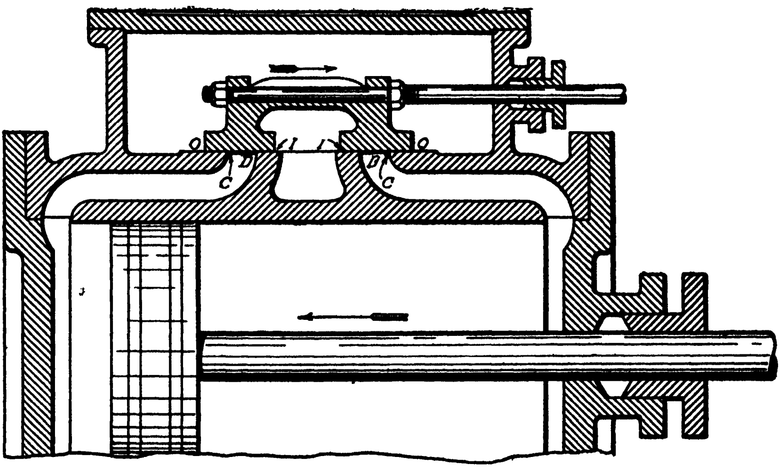


Fig. 5 Details of Cylinder, Showing Valve with Lap

placement, and both the steam and exhaust ports are wide open. The valve will not close again until the piston has reached the end of its stroke.

This type of valve is used only on small engines and, since it allows no expansion of the steam, is very uneconomical. Furthermore, it will be seen that this valve opens just after the stroke begins, which is impractical, for it means that the piston has begun its stroke before the full steam pressure reaches it, which will cause an inclined admission line on the indicator diagram.

Valve with Lap. If the face of the valve is made longer than shown in Fig. 1, so that in mid-position it overlaps the steam ports, we shall have a valve such as shown in Fig. 5. The amount that

the valve overlaps the steam ports when in mid-position is called the *lap* of the valve. In Fig. 5, $D\,I$ is the *inside lap* and $O\,C$ is the *outside lap.*

It will at once be seen that both the admission and exhaust ports may remain closed during a part of the stroke, thus making expan-

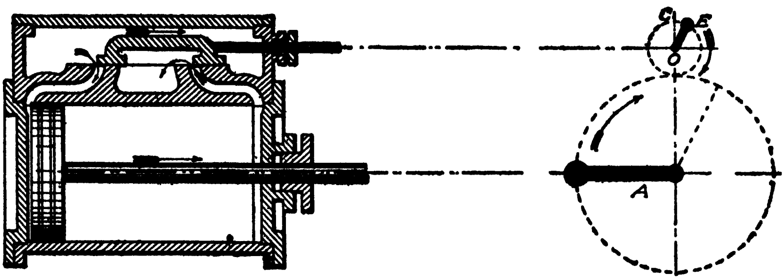


Fig 6. Valve with Inside and Outside Lap Set for Admission

sion and compression possible. It is also evident that steam can not be admitted until the valve uncovers the port by moving from mid-position a distance equal to $O\,C$. Admission continues until the valve returns to such a position that the outer edge of the valve again closes the port. Release will begin when the inner edge of the valve begins to uncover the port.

Analysis of Motion. Fig. 6 represents a valve, having both inside and outside lap, which is set at the point of admission. Since

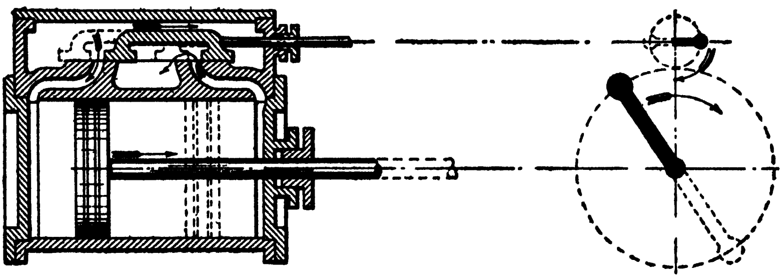


Fig 7 Valve Set at Maximum Displacement

the valve must move over a distance equal to the outside lap in order that admission may take place under proper conditions, it is evident that the eccentric can no longer be at right angles to the crank at the beginning of the stroke, but must be in advance of the

right-angle point by an amount equal to the angle EOC, known as the *angular advance.*

The maximum displacement of the valve is attained when the eccentric is horizontal, as shown in Fig. 7. In this position, both the steam and the exhaust ports are wide open, and any further

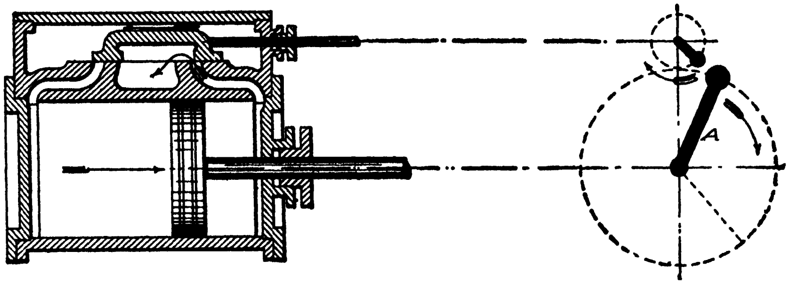


Fig. 8 Valve Position with Steam Port Closed on Head End

motion of the piston will cause the valve to move toward its mid-position.

Admission continues until the valve returns to the position shown in Fig. 8. Here the outside lap just closes the left-hand steam port, cut-off takes place, and the steam already in the cylinder begins to expand. As the valve continues to move toward the left, the left-hand inside lap begins to uncover the left-hand port and releases the steam at the position shown in Fig. 10. The dotted lines of Fig. 7 show the valve in its extreme position to the left, while the

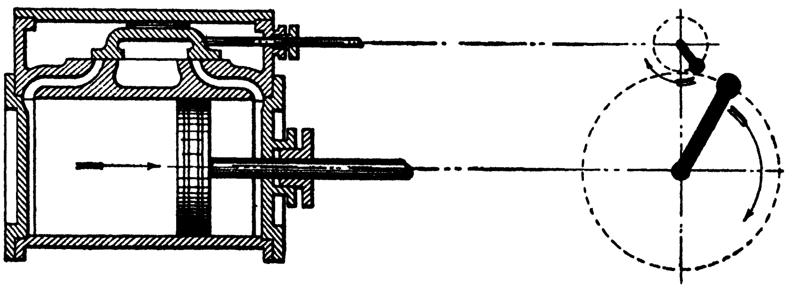


Fig 9. Valve Position with Exhaust Port Closed on Crank End

dotted position of crank and eccentric in Fig. 10 shows the valve returned to the point of compression, which continues until the conditions of Fig. 6 are again reached and the opening valve allows steam again to enter the cylinder.

This process has been traced step by step for one end only; let us now consider what is happening at the other end. While the crank A is moving from the position shown in Fig. 6 to that in Fig. 8, steam is being admitted to the head end and being exhausted from

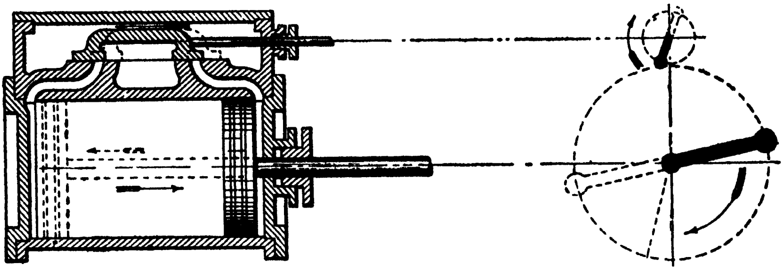


Fig. 10. Position of Valve and Cylinder for Head-End Release

the crank end. As the inside lap is less than the outside lap, the exhaust continues longer than the admission.

Fig. 9 shows the relative positions of crank, eccentric, and valve when the exhaust closes on the crank end and compression begins. Between these two positions, the steam is expanding in the head end and exhausting from the crank end. Between the positions of Figs. 9 and 10, both ports are entirely closed, and expansion is taking place in the head end and compression in the crank end. In Fig. 10 is shown the position of the valve for head-end

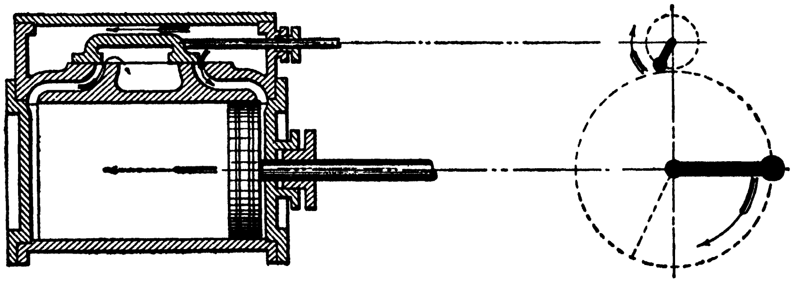


Fig 11 Position of Valve and Cylinder for Crank-End Admission

release. Fig. 11 shows admission at the crank end of the cylinder and marks the end of crank-end compression.

Effect of Change of Lap. By referring to Figs. 6 to 11, the effect of any change of lap may at once be observed. If the outside lap is increased, the valve must move farther from mid-position before

admission will occur and on the return, after the maximum displacement is reached, the greater outside lap will close the port sooner, and the point of cut-off shown in Fig. 8 will be reached before the crank reaches the angle there shown. A decrease of outside lap will make cut-off later and admission earlier.

On the other hand, if the inside lap is increased, the valve must move farther before release occurs and the crank angle will be greater than that shown in Fig. 10, while on the return to the dotted position, the port will close earlier and make an earlier compression. The crank angle will be less than is there shown. Decreasing the inside lap will cause earlier release and later compression.

Thus we see that it is the outside lap that influences admission and cut-off, and the inside lap that controls release and compression. For this reason the outside lap is often called the *steam lap* and the inside lap is called the *exhaust lap.*

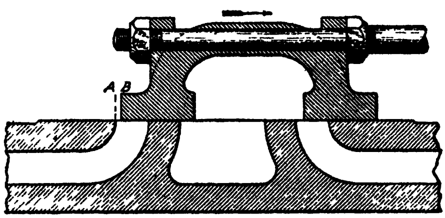


Fig. 12. Position of Valve Showing Lead

Lead. If a valve having lap is in mid-position, the port is closed and the engine can not start, because no steam can enter the cylinder. That the steam may be ready to enter the cylinder at the beginning of the stroke, it is necessary that the eccentric be set more than 90 degrees ahead of the crank as already mentioned, thus making the eccentric radius take an angular advance EOC, as shown in Fig. 6. In order that the ports and all clearance space may be properly filled with steam at the beginning of the stroke, it is necessary that the valve be displaced from its mid-position an amount slightly greater than the outside lap. With the piston at the end of the stroke, the valve will have a position as shown in Fig. 12, the port being open the distance AB, the lead of the valve. This causes the eccentric to be moved forward a slight amount in excess of the lap angle. This excess is called the *angle of lead.*

In Fig. 13, O_2R_2 represents the position of the crank at the beginning of the stroke, LO_1A_1 the lap angle, and $A_1O_1A_2$ the angle of lead. The eccentric, to give lead, must be set at the angle $R_1O_1A_2$ ahead of the crank or 90 degrees plus the angular advance. In large

high-speed engines, a liberal lead is essential in order that the ports and clearance space may be well filled with steam before the stroke begins. If there is no lead, a portion of the steam will be used in filling these places and full steam pressure will not reach the piston until it is well advanced on the stroke. This will give a sloping admission line, as shown in Fig. 14. Too much lead, on the other hand, will cause too early an admission, as shown in Fig. 15.

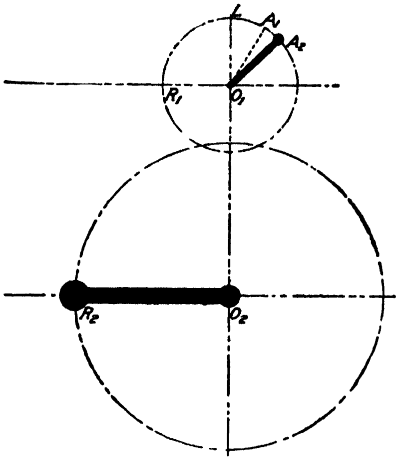


Fig. 13 Diagram Showing Lap Angle and Angle of Lead

If the angular advance is increased, the eccentric will be moved further ahead of the crank, and consequently it will arrive at each of the events sooner than before. If, then, the angular advance is increased, all of the events of the stroke will occur earlier.

Effect of Lead. From the foregoing discussion of lead, it is evident that its effect is to permit steam to enter the cylinder before the end of the stroke, which tends to provide an abundance of steam behind the piston when starting the return stroke and throughout the period of admission. It also promotes smooth running of the engine by furnishing a cushion or retarding force to the moving parts, thereby eliminating the "knocks" or "pounds" incident to lost motion. Since the effect of lost motion depends upon the weight

Fig 14. Indicator Diagram Showing Effect of No Lead

Fig. 15. Indicator Diagram Showing Too Early Admission

and velocity of the reciprocating parts, it is evident that the amount of lead required will vary for different engines and for the same engine running at different speeds. The exact amount of lead can not be determined except by trial and by use of the steam engine indicator.

When experimenting for the determination of the proper amount of lead for a specific case, it will be necessary to gradually increase the angular advance until smooth running is obtained. After this result is obtained, indicator cards should be taken to see if the lead is excessive, in which case the valve must be adjusted until the desired conditions are obtained. Since lead permits steam to act against the piston before the end of the stroke, it results in negative work, hence the amount of lead should not be excessive. An amount of lead sufficient to insure the filling of the clearance space is permissible, but very much more than this is detrimental to the economic performance of the engine.

Analytical Summary of Valve Terms. Thus far in discussing the plain slide valve, a number of terms have been used that are of primary importance and must be thoroughly understood in order to properly grasp much that is yet to be studied. It seems advisable, therefore, that a recapitulation of the terms used be presented.

Mid-Position. A valve is said to be in mid-position when the center of the valve and valve seat coincide. When in this position, the steam ports are all closed.

Displacement. The displacement or a valve is the amount the valve has been moved either to the right or left of its mid-position. In Fig. 4, the valve has moved to the right a distance equal to the width of the steam port, hence in this instance the displacement of the valve is equal to the width of the steam port.

Valve Travel. The travel of the valve is the distance the valve travels in moving from one extreme position to the other. The travel of the valve is twice the eccentricity, or throw of the eccentric.

Eccentricity. The eccentricity, or throw of the eccentric, is the distance between the center of the shaft and the center of the eccentric. It is equivalent to a crank, the length of which is one-half the valve travel. For instance, if the valve travel of an engine is 6 inches, the eccentricity, or throw of the eccentric, would be 3 inches, or one-half of the valve travel.

Lap. The amount that the valve extends over the steam port when in mid-position is called steam lap or often spoken of as the lap of the valve. The steam lap is equal to $O\,C$ in Fig. 5. In Fig. 5, it is obvious that when the valve is in mid-position, the distance $D\,I$ is called exhaust lap. Steam lap and exhaust lap are frequently

spoken of as outside and inside lap, respectively. The effect of the exhaust lap is to delay exhaust and hasten compression.

Very frequently a valve does not have any exhaust lap and there is a small port opening between the cylinder and the exhaust cavity when the valve is in mid-position, as shown at *A*, Fig. 19. In such a case, the valve is said to have *inside clearance.* The effect of inside clearance is opposite to that of exhaust lap, namely, it delays compression and hastens exhaust, and insures a minimum amount of back pressure.

Lead. By the term *lead* is meant the amount the steam port is open when the engine is on either dead center.

Angle of Advance. It was noted in Fig. 1 that the crank and eccentric were exactly 90 degrees apart and that admission occurred at the beginning and cut-off at the end of the stroke. On account of economic reasons, this is not a good arrangement. Hence we find that the valves on all engines have lap and are set to give the necessary amount of lead. In order to obtain lead when the engine is on dead center with a valve having lap, it is necessary to turn the eccentric ahead, in the direction the engine is to run, through such an angle that the valve will be displaced by an amount equal to the lap plus the lead. The angle measuring this displacement is the sum of the angle of lap and the angle of lead. If there is no lead, this angle would be decreased by the angle of lead. The sum of the angle of lap and the angle of lead is frequently designated as the angle of advance. The angularity between the eccentric and the crank then becomes equal to 90 degrees, plus or minus the angle of advance according to the type of valve and gear.

Inequality of Steam Distribution. In the valve diagrams thus far considered, the events of the stroke have been discussed for each end separately, without reference to the relation of similar events on the other side of the piston. If the connecting rod were of infinite length, so that it would always remain parallel to the center line of the engine, the distribution would be the same for both ends of the cylinder. In practice, the connecting rod varies from four to eight times the length of the crank, which causes the connecting rod always to be at an angle to the center line of the engine when the engine is off dead center, and for a given crank angle makes the piston displacement greater at the head end than at the crank end.

To Find Displacement of Valve. The circle, Fig. 16, represents the path of the eccentric center during a complete revolution of the engine. OC represents the crank, and OR the corresponding position of the eccentric. The diameter XY represents the extent of the valve travel. Since the eccentric rod is so long in comparison to the eccentricity, we make no appreciable error by assuming it always to be parallel to the center line of the engine. When the eccentric is at OL, the valve is in mid-position. At OR the valve has moved from mid-position an amount ON, found by dropping a perpendicular from R to the center line XY. If the angularity of the connecting rod could be neglected, the piston displacement could be found in the same manner.

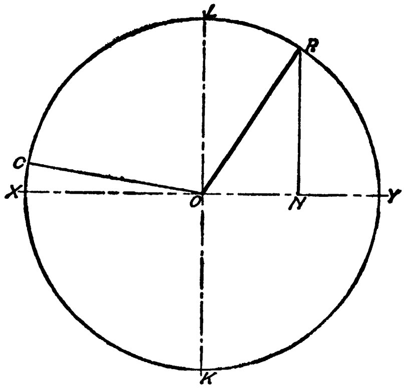


Fig 16 Eccentric Circle Showing Relative Positions of Crank and Eccentric

To Find Displacement of Piston. To find the displacement of the piston, a diagram as shown in Fig. 17 must be drawn. In this figure, AB represents the cylinder, P_1 the piston, H_1 the crosshead, H_1R the connecting rod, and OR the crank. Suppose now the engine should stop and the piston be clamped in this position. The piston displacement would be represented by AP_1. If the crank pin at R should now be loosened so as to allow the connecting rod to fall to a

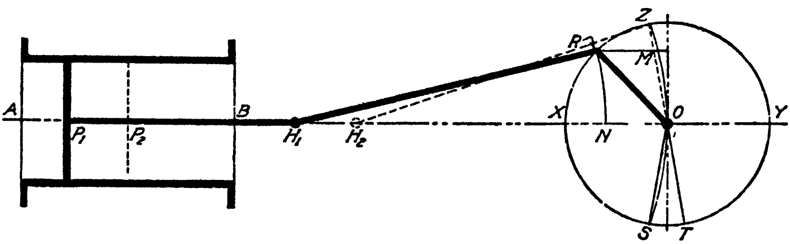


Fig 17 Diagram for Finding Displacement of Piston

horizontal position, the point R would describe the arc of a circle RN, and XN would represent the piston displacement and would be equal to AP_1. Suppose now that in this disconnected way, the

piston, crosshead, and connecting rod were moved forward until the end of the rod came to O.' P_1 would then be at P_2 and the piston would be in the middle of its stroke. Now suppose the end of the rod were swung up to its proper position on the crank-pin circle, the piston remaining stationary. The end of the rod would describe an arc OZ; the crank pin would be at Z, less than a quarter revolution from X; while the piston would be in the middle of its stroke.

Suppose this engine were running with cut-off at half stroke on the head end, and that XOZ represented the corresponding crank angle. On the return stroke, the valve would cut off at the same crank angle YOT, which is equal to XOZ, and OT would represent the crank position for cut-off on the return, or crank-end, stroke. The piston, as we have just seen, will not be at half stroke except

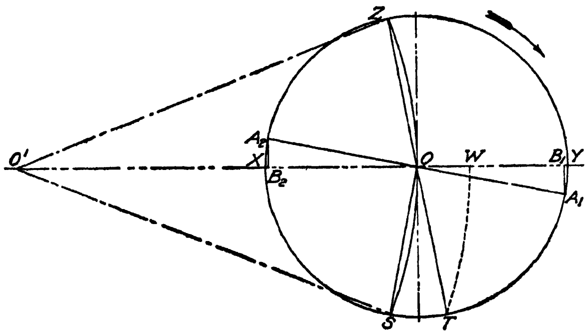


Fig 18. Crank and Eccentric Diagram for Engine Shown in Fig 17

when the crank is at OZ or OS. Consequently, the crank position OT is less than half stroke and cut-off occurs earlier at crank end than at head end. When the crank is at OZ, the eccentric will be at OA_1, Fig. 18, assuming the valve to have no lap, and the valve displacement will be OB_1. When the crank is at OT, the eccentric will be at OA_2 and the valve displacement will be OB_2, which is equal to OB_1, the displacement of the valve at cut-off on the head end. The piston displacement will be OX on the head end and WY on the crank end when cut-off occurs. If the connecting rod always remained parallel to the center line, the cut-off would be the same at both ends.

Compensation of Cut-Off. It has been pointed out that lengthening the outside lap makes the cut-off earlier, and that shortening

the lap makes it later. The cut-off in the case just cited may then be equalized by altering the outside laps. If we increase the outside lap on the head end, or decrease the crank-end lap, the inequality will be less. By changing either or both of the laps the proper amount, the cut-off may be exactly equalized.

But altering the outside lap changes the lead, as has already been explained. If the lap is increased on the head end, the lead will be less than on the crank end. If the lead becomes too small on the head end, the angular advance may be increased but the inequality of lead will still remain, for this increase of angular advance will increase the lead at the crank end as well as at the head end, and by hastening all the events of the stroke may give a bad steam distribution if the proper care is not taken.

Unequal lead is of less consequence on a low-speed engine than on a high-speed engine. On low-speed engines, the cut-off may be

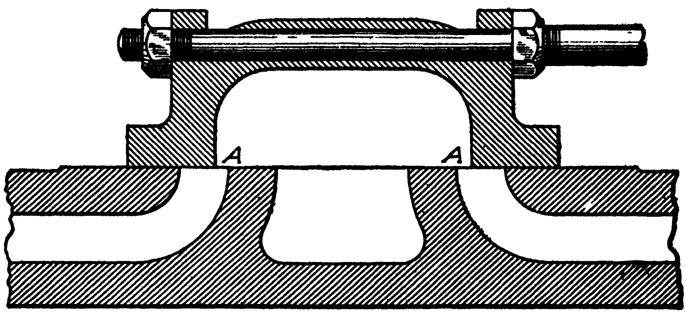


Fig 19. Valve in Mid-Position Showing Inside Clearence, or Negative Lap

equalized at the expense of lead with beneficial results, but on high-speed engines, this is not true. A high-speed engine requires more lead than a low-speed engine, for there is relatively less time in each stroke for the clearance space to fill with steam.

If both inside laps are equal, compression will not occur equally at both ends. To equalize it, the inside laps may be changed in the same manner as the outside laps are changed to equalize the cut-off. By altering these inside laps to equalize compression, it may happen that the lap is reduced enough to leave the exhaust port open when the valve is in mid-position. The amount of this opening is called *inside clearance*, or *negative lap*. This is illustrated at A, Fig. 19.

Rocker. Sometimes it happens that the valve stem and eccentric rod can not be so placed that they will be in the same straight line; or it may be that the travel of the valve must be so great as to require an excessively large eccentric. In such cases, a rocker may be used.

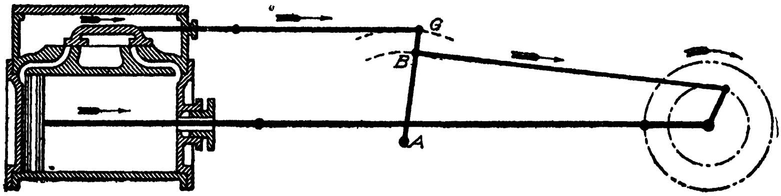


Fig. 20 Valve with Rocker Arrangement

Fig. 20 shows a valve that is not in line with the eccentric. An instance where this occurs is in horizontal engines when the valve is set on top of the cylinder instead of on one side. By means of the rocker *A G*, the valve may receive its proper motion.

In case it is more convenient to place the pivot of the rocker arm between the connections to the valve stem and those of the eccentric rod, such an arrangement as is shown in Fig. 21 may be used. Here it will be noticed that the valve stem and eccentric rod are moving in opposite directions and in order to give the valve the same motion as in Fig. 20, the eccentric must be moved 180 degrees ahead of the position there shown.

If *A B* is less than *A G*, the valve travel will be greater than twice the eccentricity, in proportion as *A G* is greater than *A B*. In all

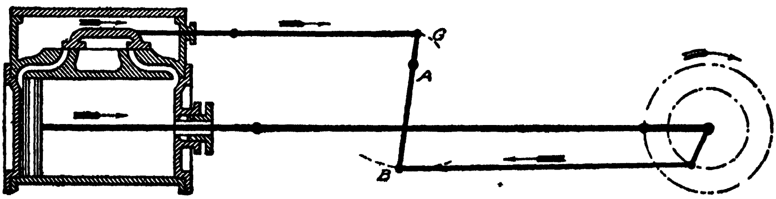


Fig 21 Arrangement of Rocker by which Valve Stem and Eccentric Rod Move in Opposite Directions

cases, the valve travel is in the same proportion to twice the eccentricity as *A G* is to *A B*. Thus, if the valve travel is $4\frac{1}{2}$ inches, *A B* is 15 inches, and *A G* is 18 inches, then $\frac{15}{18}\times4\frac{1}{2}=3\frac{3}{4}$ inches, will equal twice the eccentricity.

A valve gear may be so laid out as to make both the cut-off and the lead equal for both ends of the cylinder. This may be done by

a proper proportion between the rocker arms, and a careful location of the pivot of the rocker. The eccentric must then be set accordingly. In this manner, the Straight Line engine equalizes the cut-off and lead. A discussion of this method will be considered later.

VALVE DIAGRAMS

Zeuner Diagrams. In order to study the movements of the valves, the effects of lap, lead, eccentricity, etc., diagrams of various sorts have been devised. By the use of diagrams we may acquire

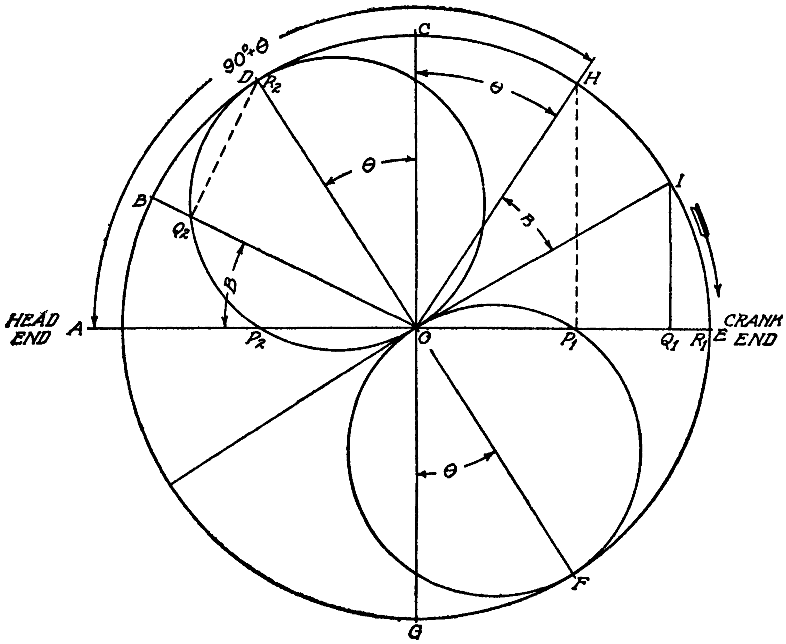


Fig 22. Zeuner Diagram for Valve Analysis

a knowledge of the valve motion without the complex mathematical expressions that such a discussion would entail. The most useful of these various diagrams is that devised by Zeuner and, to avoid complexity, we shall confine ourselves to a discussion of this diagram alone. The eccentric rod is assumed to be of infinite length, and the positions of the crank are shown on the diagrams. The displacement of the piston can easily be found if the ratio of crank to connecting rod is known.

The function of the Zeuner diagram is to show the relation between the valve positions and crank positions. This relation being known, it is a simple matter to obtain the eccentric and piston positions.

In Fig. 22, $A\,O\,E$ represents the valve travel, and the center of the eccentric will move in the circle $A\,C\,E\,G$. It is assumed, also, that this circle represents the path of the crank center, hence this circle will be known as the *crank and valve circle*. OA is the position of the crank and $O\,H$ is the corresponding position of the eccentric, when the engine is on the head-end dead center. Since this valve has lap, and since admission must occur before the end of the stroke, it is evident that the eccentric must precede the crank by 90 degrees plus the angle of advance θ. From H drop a perpendicular line upon the center line $A\,O\,E$, thus locating the point P_1. The distance OP_1 is the amount the valve has been moved to the right of its mid-position when the crank is on dead center. Since the diagram gives the relation between crank and valve positions, the displacement of the valve $O\,P_1$ can be laid off from O on the crank position OA, thus establishing the point P_2. Turn the crank through an angle B to the position $O\,B$. The eccentric will move through the same angle and will be found at I. Draw the perpendicular line IQ_1, and OQ_1 represents the displacement of the valve for the crank position $O\,B$. Lay off OQ_1 on $O\,B$, establishing the point Q_2. Continue the rotation of the crank until the point D is reached. The eccentric then will be found at E, and the valve will have its greatest displacement $O\,R_1$ to the right of its mid-position. It is evident that $O\,R_1$ is equal to $O\,D$. If the rotation of the crank be continued in the direction of the arrow, the valve will return from its extreme position on the right and will approach its mid-position. By locating on the various crank positions the corresponding valve displacement, a series of points as P_2, Q_2, R_2, etc., will be obtained, all of which will lie on the circumference of a circle, as $OP_2Q_2R_2$, the diameter $O\,D$ of which will make an angle θ equal to the angle of advance laid off to the left of the vertical $O\,C$. If this operation be continued for a complete revolution, a series of points will be established in the lower quadrant, establishing a circle OP_1F, the diameter of which will be a continuation of $O\,D$ and, therefore, will make an angle θ with the vertical but will lie on the right of

the vertical line COG. These two circles are called *valve circles*, and they represent the movement of the valve to the right and left of its mid-position and, as previously stated, represent the amount the valve has moved for any crank position such as OB.

Having established the valve circles, it is a simple matter to obtain the valve displacement for the position OB, which, in this case, would be the distance OQ_2 cut off from OB by the valve circle. It can be proven that OQ_2 is the valve displacement by comparing the two right triangles OIQ_1 and ODQ_2. They are equal because

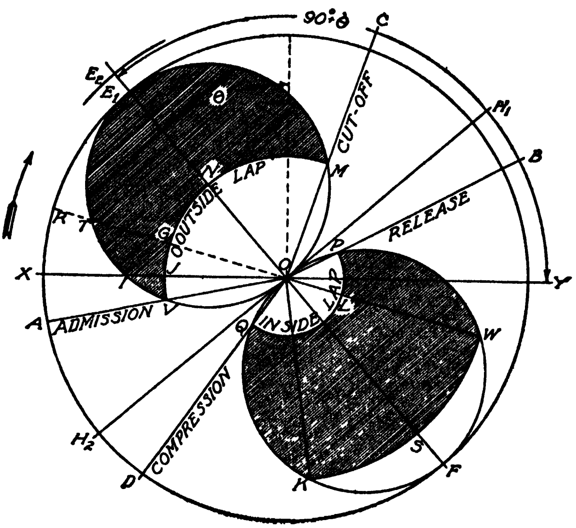


Fig 23. Diagram Showing Study of Valve Motion for Head End Only

they are similar and have two corresponding sides OD and OI equal. Therefore, OQ_2 equals OQ_1. This being true for any crank position, it is true for all crank positions.

Study of Valve Motion from Diagram. Now that the truth of our proposition has been proved, let us see how we may study the valve motion from such a diagram. In Fig. 23 let XY represent the valve travel; then the circle XE_1YF will represent the path of the center of the eccentric. Let θ be the angle of advance and lay off E_1O toward the crank, making an angle θ with the vertical. Produce E_1O to F, and on OE_1 and OF as diameters draw the valve circles as shown. With O as a center and OV, equal the outside lap, as a

radius, draw an arc intersecting the upper valve circle at V and M. Lay off $O P$ equal to the inside lap and with O as a center and $O P$ as a radius, draw an arc intersecting the valve circle at P and Q. Draw the crank-position line $A O$ passing through V. Then, when the crank is in this position, the displacement of the valve is equal to $O V$ (the outside lap) and the steam is ready to enter the cylinder. This is the position of the crank at admission, and the crank angle XOA is called the *lead angle*. The valve has lead and, therefore, the admission takes place before the end of the stroke. When the crank reaches the position OE_1, the displacement of the valve is equal to the eccentricity OE_1, and is at a maximum. Further motion of the piston causes the valve to move toward mid-position until, at the crank position OC, the displacement OM is again equal to the outside lap and the valve has reached the point of cut-off. When the position OH_1 is reached, the crank line is tangent to both valve circles and there is no displacement of the valve. At this point, the valve is in mid-position.

Further crank movement draws the inside lap toward the edge of the exhaust port until, at the crank position $O B$, the displacement is equal to OP (the inside lap) and release begins. At OF the maximum valve displacement is again reached and the valve moves in the opposite direction until at OD its displacement from mid-position is again equal to $O Q$, equals OP the inside lap, and compression takes place. At OH_2 the valve is again in mid-position. At OX the displacement of the valve is OI, but since the valve has to move the distance OJ before the port begins to open, IJ must represent the port opening when the crank is on dead center, and by definition we know that lead is the amount of port opening at this position. Therefore, $I J$ represents the lead.

At the position R, the port is open an amount equal to TG; at E_1 the opening is a maximum equal to E_1N; at C the port is on the point of closing and there is no opening. If LW represents the total width of the steam port, the exhaust port will be open wide when the displacement of the valve is equal to $O W$ and it will remain wide open while the crank swings from OW to OK.

If the width of steam port in addition to the outside lap were laid off on the other valve circle, it would fall at E_2. For the admission port to be wide open, the displacement of the valve would have

to be equal to OE_2 which is more than the maximum displacement. This shows that in this case the steam port is never fully open and that the left-hand edge of the valve overlaps the right-hand edge of the port by an amount equal to E_1E_2 when the valve has reached its maximum displacement.

Fig. 23, with its two valve circles, shows the diagram for the head end of the cylinder only. The crank-end diagram would be similar except that the laps might not be equal to those of the head end.

Properties of Zeuner Diagrams. The Zeuner diagram deals with admission, cut-off, release, compression, lead, valve travel, angle of

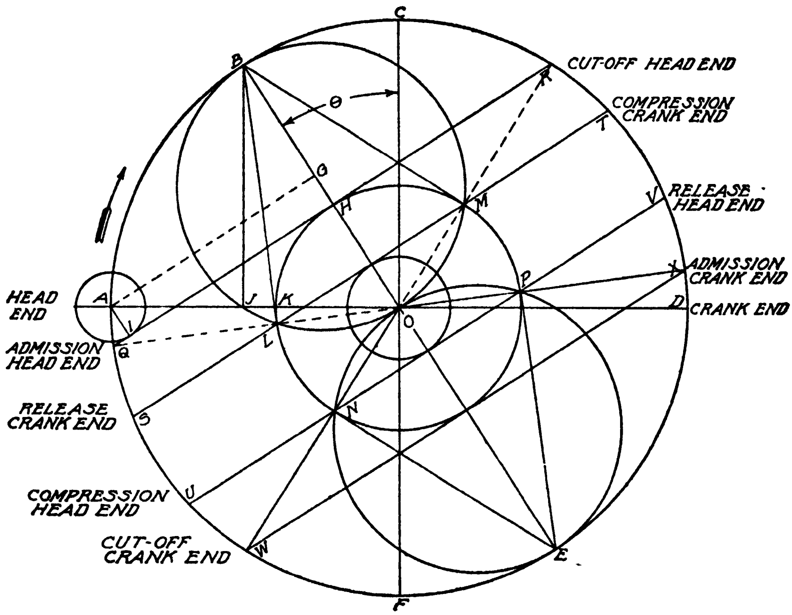


Fig. 24. Diagram Analysis for Movement of a Direct Valve as Regards Head End of Cylinder

advance, maximum and minimum port opening, steam lap, and exhaust lap. Generally, if four of these be given, the others can be found. It is evident, therefore, that there are a great many possible combinations, hence it is necessary to have definitely in mind and clearly understood the properties of the Zeuner diagram. The proofs given are for the movement of a direct valve as regards the head end of the cylinder. All letters refer to Fig. 24.

(1) The figure is symmetrical on the line BE. In the semicircles OLB and OMB, OL equals OM, each being the radius of the steam-lap circle. Since OL equals OM, the arcs which they subtend are equal, therefore, the arcs LJB and MB are equal. This makes the angles LOB and MOB equal because they are measured by equal arcs. Therefore, BO bisects the angle LOM, and in a similar way it can be proved that OE bisects the angle NOP.

(2) The line BM is perpendicular to OMR and is tangent to the steam-lap circle. The angle BMO is a right angle because it is inscribed in a semicircle. Therefore, BM is tangent to the steam-lap circle and is perpendicular to the crank position OMR.

(3) The line joining the admission and cut-off points for the head end is perpendicular to BO and is tangent to the steam-lap circle.

The triangle QOR is an isosceles triangle and, as demonstrated above, BO bisects the angle QOR, hence BO is perpendicular to the base QR. To prove that QR is tangent to the steam-lap circle, it is necessary to show that the distance OH measured on BO is equal to OM, the radius of the steam-lap circle. The right triangles BOM and HOR are equal, having two sides equal and one common angle. Hence, OH is equal to OM.

(4) The line BJ is perpendicular to AO. The angle BJO is a right angle, being inscribed in a semicircle.

(5) The radius of the circle AI with center at A and tangent to QR, is equal to the lead JK.

From the center A draw AG parallel to IH. In the right triangles BJO and AGO, the angle AGO equals BJO, being right angles. BO equals AO. The angle AOH is common to both triangles, therefore, they are equal. Hence, OJ equals OG. But OK equals OH. Therefore, GH equals JK, equals AI, which is the lead.

By using Fig. 24 at all times as a reference figure and bearing in mind the things it tells, no great difficulty should be encountered in solving problems. To illustrate the principles set forth above and to give an idea of the practical use of the Zeuner diagram, several problems will be worked out as an indication of what may be done.

ILLUSTRATIVE PROBLEMS

In designing a slide valve, a few of these variables depend upon the conditions under which the engine is to run. For instance, the valve travel is limited, cut-off must be at a certain point, and the engine must have a certain lead. Then, with the aid of a Zeuner diagram, the remaining proportions of the valve may be determined.

EXAMPLE 1. Given a valve travel of 3 inches, exhaust lap of $\frac{3}{4}$ inch, angular advance of 35 degrees, and crank angle at cut-off of 115 degrees Determine the laps, the lead, and the crank angles at admission, compression, and release

SOLUTION. In Fig. 25, let *X Y* represent the valve travel of 3 inches. Draw *O M* perpendicular to *X Y*, and on *X Y* as a diameter draw the circle

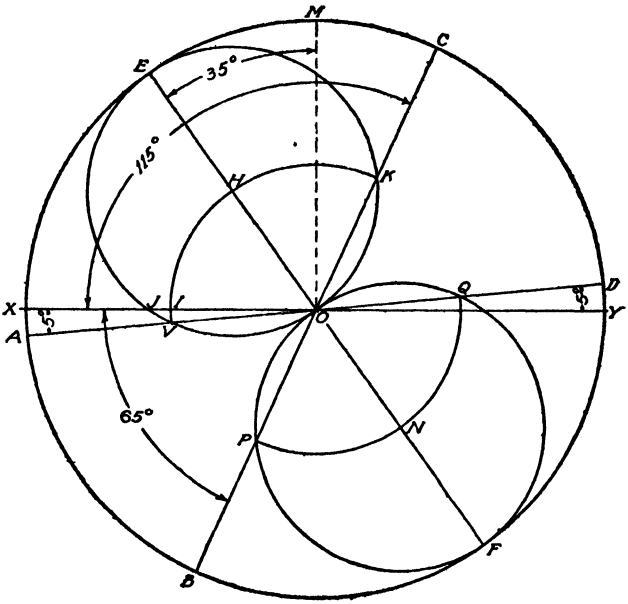


Fig 25. Zeuner Diagram for Finding Laps, Lead, and Crank Angles

X M Y F representing the path of the center of the eccentric as it revolves about the shaft. Lay off the angle *M O E* to represent the angular advance of 35 degrees so that the angle *X O E* is equal to 90 degrees minus the angular advance. Produce *E O* to *F*. Then on *O E* and *O F* as diameters, draw the valve circles. The eccentricity *O E* or *O F*, if no rocker is used, will be half the valve travel. Lay off the angle *X O C* to represent the crank angle at cut-off of 115 degrees, and *O K* will then represent the distance of the valve from mid-position when cut-off takes place. This distance we know is the outside lap. Draw the arc *K I*, known as the lap circle, and it will cut the valve circle again at *V*. When the valve is again the distance *O V*, the out-

side lap from mid-position, admission will take place. Draw the line $O\ V\ A$ and this will represent the position of the crank at admission.

When the crank is at $O\ X$, the valve displacement is equal to $O\ J$. This is at dead center, and the valve is open the amount $I\ J$, for it has moved this distance more than the outside lap. Therefore, $I\ J$ is the lead for this end.

Now on the other valve circle, draw the arc $P\ Q$ with the inside lap ($\frac{3}{4}$ inch) as a radius. It will cut the valve circle at P and Q. When the valve displacement is equal to $O\ Q$, the exhaust port has just commenced to open, and the engine is at release In the same way, when the valve displacement is equal to $O\ P$, the port begins to close and the engine is at compression $O\ Q\ D$ represents the crank position at release and $O\ P\ B$ the crank position at compression

The results are tabulated as follows:

Outside lap $O\ K$	$=\frac{3}{4}$ inch
Angle of lead $X\ O\ A$	$=5$ degrees
Linear lead $I\ J$	$=\frac{3}{32}$ inch
Max port opening for admission $H\ E$	$=\frac{3}{4}$ inch
Crank angle at release $X\ O\ D$	$=185$ degrees
Crank angle at compression $X\ O\ B$	$=$ 65 degrees
Max. port opening for exhaust $F\ N$	$=\frac{3}{4}$ inch

Fig. 25 is drawn full size, and all of the above measurements may readily be verified. This figure is drawn for the head end only If the crank angle at cut-off is the same on both ends, the Zeuner diagram for the crank end will be exactly like Fig. 25.

EXAMPLE 2. Given a lead $\frac{1}{16}$ inch, valve travel 3 inches, steam lap (h.e. and c.e) $\frac{7}{8}$ inch, exhaust lap (h e and c e) $\frac{3}{16}$ inch. Let $\frac{R}{L}$, that is, the ratio of the length of the crank to the connecting rod, equal $\frac{1}{5}$ Construct the Zeuner diagram and find all the events for both the head and crank ends in per cents.

SOLUTION Construct the valve travel circle $A\ C\ D\ F$, Fig 26, with a radius of $1\frac{1}{2}$ inches; the steam-lap circle with a radius $O\ H$ of $\frac{7}{8}$ inch; and the exhaust lap circle with a radius $O\ R$ of $\frac{3}{16}$ inch The steam-lap circle cuts the crank position for h.e. dead center at the point K. From K lay off the distance $J\ K$ to represent the lead of $\frac{1}{16}$ inch At A, construct the lead circle with a radius of $\frac{1}{16}$ inch. From the properties of the Zeuner, we know that where a perpendicular erected at the lead point J cuts the valve travel circle as at B, the line $B\ O$ is the diameter of the valve circle and the angle $C\ O\ B$ is the required angle of advance. We also know that a line drawn perpendicular to $B\ O$ and tangent to the steam-lap circle cuts the valve travel circle at the points of admission and cut-off, respectively Therefore, draw $S\ T_2$ so it will be tangent to the steam-lap circle and perpendicular to $B\ O$ at H The points S and T_2 are the points of head-end admission and cut-off, respectively. It is to be noted, also, that this line $S\ T_2$ is tangent to the lead circle, which fulfills another condition of the property of the Zeuner.

To locate the other events for the head and crank ends, draw lines perpendicular to $B\ O\ E$ and tangent to the steam- and exhaust-lap circles, and the points where these lines cut the valve travel circles will be the required

points. In the same manner, the several other points in the figure have been located.

To find the per cent of stroke at which the several events occur, take a radius proportionately equal to the length of the connecting rod and describe the arcs shown. As $\frac{R}{L}$ is $\frac{1}{5}$, L equals 5 R. But R is one-half the valve travel, *i e.*, 1.5 inches.

$$\therefore \quad L=5\times1.5$$
$$=7.5 \text{ inches}$$

Now, with a radius of 7 5 inches and with a center on the horizontal line through the center of the valve travel circle produced to the left of the

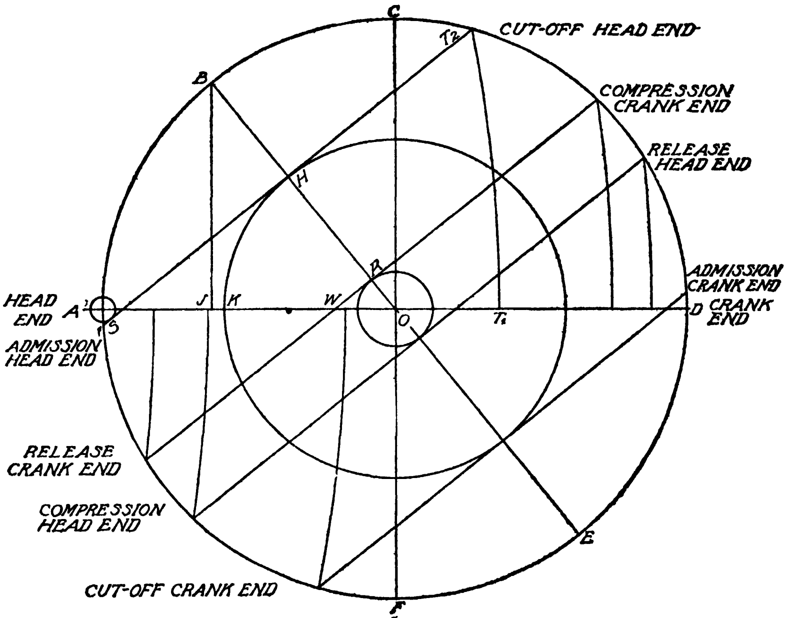


Fig 26 Diagram for Finding Events for Head and Crank Ends, Lead, Valve Travel, and Laps being Given

vertical line $C F$, sweep the arcs shown from the points of admission, cut-off, etc., on the head and crank ends Remembering that the head-end events are measured from the head-end dead center and the crank-end events from the crank-end dead center, measure the distance $A\ T_1$. This distance, 2.03 inches, divided by the valve travel 3 inches, and multiplied by 100, gives the per cent cut-off on the head end, that is,

$$\frac{2.03}{3}\times 100=67\% \text{ cut-off}_{\text{h e.}}$$

In like manner, measure the distance $D W$ for the crank-end cut-off, which we find is 1 8 inches. Then

$$\frac{1\,8}{3}\times 100 = 60\% \text{ cut-off}_{\text{h.e.}}$$

Continuing this procedure for the other events, the final results obtained from the diagram will be

Event	Head End	Crank End
Admission	98 per cent	98 per cent
Cut-off	67 per cent	60 per cent
Release	93 per cent	91 per cent
Compression	16 per cent	12½ per cent

Angle of advance $\theta = 40$ degrees

Example 3 Given an engine having 30 per cent cut-off on the head end; maximum port opening of ⅝ inch; and lead on the head end $\frac{1}{16}$ inch. The laps are to be equal; compression on the head end is 25 per cent; and $\frac{R}{L}$ equals ¼ Construct the Zeuner diagram

Solution. In Fig. 27, lay off $E\,F$ to represent the maximum port opening ⅝ inch; $F\,G$ the lead $\frac{1}{16}$ inch; and erect perpendiculars $E\,J$, $G\,H$, $F\,I$. On any point as O on the line GH, draw a trial circle such as $A\,B\,C\,D$, which in this case was assumed to be 1 inch in diameter Since cut-off on the head end occurs at 30 per cent of the stroke, locate the direction of the crank position $O\,P$ for this position. This direction will hold for any valve travel Draw $O\,M$ perpendicular to $O\,P$, cutting $F\,I$ at K. Bisect the angle $F\,K\,M$ by $K\,N$. On $K\,N$ as a center line, find by trial a radius and center, such that a circle when described will pass through O and be tangent to $E\,J$. The center is found to be at L and the distance $O\,L$ is the radius of the required valve circle. With L as a center, draw a circle tangent to $F\,I$ and $K\,M$. Such a circle will be the required steam-lap circle. To demonstrate why this construction is correct, it is only necessary to refer to the properties of the Zeuner diagram as given in connection with Fig 24. Here it is shown that a line drawn perpendicular to the crank position for the point of cut-off and tangent to the steam-lap circle cuts the valve travel circle at the extremity of the valve circle, as at B Hence, $O\,M$ fulfills this condition, which gives the extremity of the required valve travel circle at O. In Fig. 24, it is also evident that the steam-lap circle is tangent to the perpendicular to the crank position for the given cut-off and is also tangent to a perpendicular to the horizontal center line drawn at the extremity of the maximum port opening. Therefore, this condition was fulfilled in establishing the required lap in Fig. 27. Having obtained the valve travel and lap, it remains to complete the diagram in order to determine the other conditions. In Fig. 28, the circles $A\,B\,C\,D$ and $a\,b\,c\,d$ are constructed on a diameter of $4\frac{11}{16}$ inches and $4\frac{1}{16}$ inches, respectively, the former being the value of the valve travel and the latter twice the steam lap, as found in Fig. 27. Locate the head-end cut-off at 30 per cent and draw the lead circle with a radius of $\frac{1}{16}$ inch. Locate the head-end compression of 25 per cent at I. Draw $G\,H$ tangent to the

steam-lap and lead circles cutting the valve travel circle at G, thus establishing the head-end admission. From the properties of the Zeuner diagram as discussed on pages 21 and 22, we know the diameter of the valve circles will be on a line bisecting the angle GOH. Draw the line $F\,O\,E$ bisecting this angle; this line will be perpendicular to $G\,H$. Having established $F\,O\,E$ and bear-

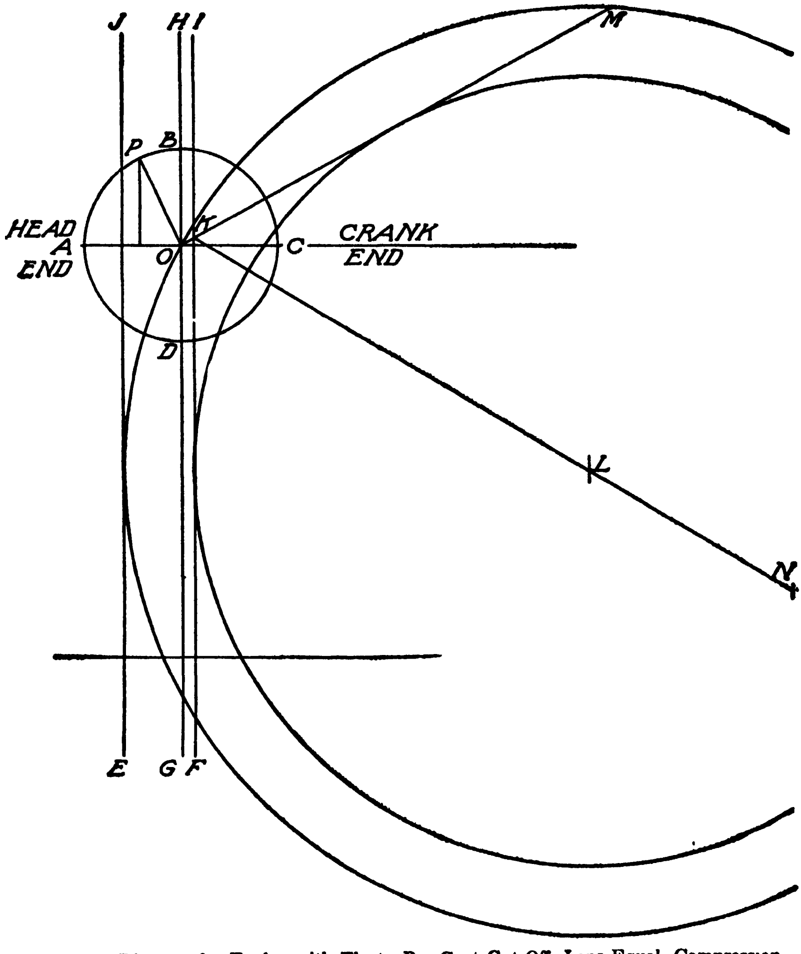


Fig. 27. Diagram for Engine with Thirty Per Cent Cut-Off, Laps Equal, Compression Twenty-Five Per Cent

ing in mind the demonstrations previously given, it is evident that a line drawn from the point of compression on the head end I perpendicular to $F\,O\,E$ will cut the valve travel circle at J, the point of head-end release. It is to be noted, however, that the line joining the points of release and compression on the head end lies on the same side of the center O as does the line joining the points

of admission and cut-off for the head end. This relation being opposite to that found in Fig. 24 means that instead of having exhaust lap with this valve, there is inside clearance equal to $O N$. With O as a center and $O N$ as a radius, describe the clearance circle and complete the Zeuner by drawing the parallel lines $K L$ and $P Q$, thus locating the remaining events of the stroke. In order to obtain the per cents of the events of the stroke, proceed as in Example 2

The results are tabulated as follows:

Steam lap	$=2\frac{1}{32}$ inches
Inside clearance	$=\frac{5}{16}$ inch
Valve travel	$=4\frac{13}{16}$ inches
Angle of advance	=62 degrees
Admission on both h.e. and c.e.	=99 per cent (approx.)
Cut-off c.e.	=20 per cent
Compression c.e.	=18 per cent
Release c.e.	=60 per cent
Release h.e.	=72 per cent

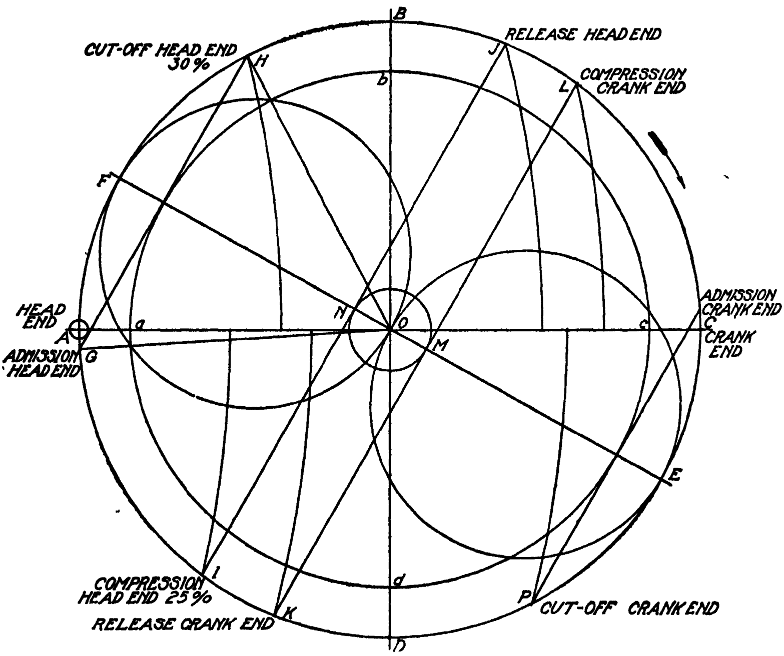


Fig. 28 Diagram for Example 3 to Determine Admission, Compression, and Release at Crank and Head Ends

The preceding problems involve nearly all of the properties of the Zeuner diagram and, if completely mastered by the student, should make the solution of other problems very much easier.

Effect of Changing Lap, Travel, or Angular Advance. We are now in a position to consider more in detail the effect of changing

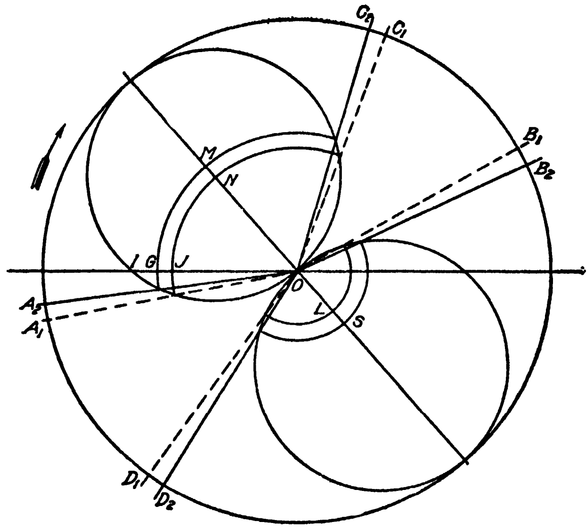


Fig 29 Study of Effect of Changing Valve or its Setting

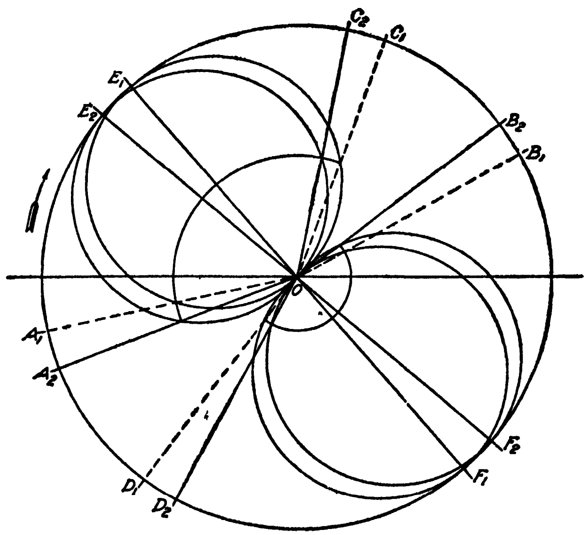


Fig 30. Study of Effect of Changing Angle of Advance

in any way either the valve or the setting. Let us consider Fig. 29, which is in every way like Fig. 23 except that all unnecessary

TABLE I

Effect of Changing Lap, Travel, and Angular Advance

Event	Increasing Outside Lap	Increasing Inside Lap	Increasing Travel	Increasing Angular Advance
Admission	Is later Ceases sooner	Not changed	Begins earlier Continues longer	Begins earlier Same period
Expansion	Is earlier Continues longer	Beginning unchanged Continues longer	Begins later Ceases sooner	Begins earlier Same period
Exhaust	Unchanged	Occurs later Ceases sooner	Begins earlier Ceases later	Begins earlier Same period
Compression	Begins at same point Continues longer	Begins sooner Continues longer	Begins later Ceases sooner	Begins earlier Same period

letters and lines are omitted to avoid confusion. If the outside lap, or steam lap, is *increased* an amount equal to $N M$, the admission will take place later, viz, at crank position $O A_2$; the lead will be reduced to $I G$ and cut-off will take place earlier, viz, at $O C_2$.

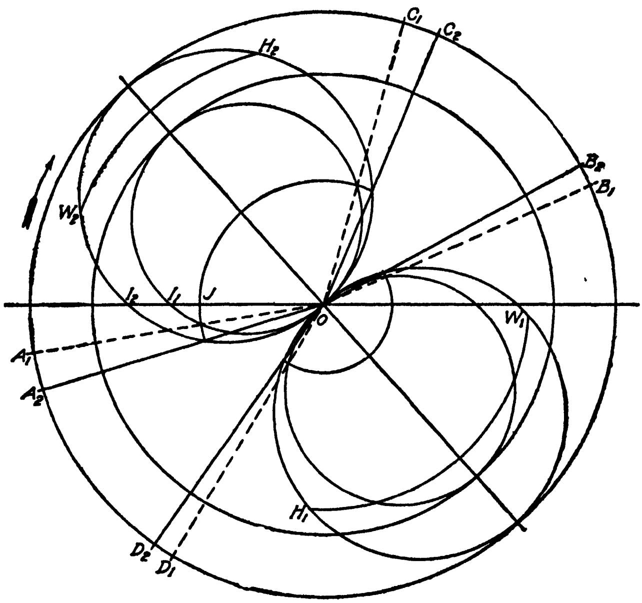


Fig. 31 Study of Effect of Changing Eccentricity

If the outside, or steam lap, is *reduced* a like amount, the contrary effects will be observed. If the inside lap, or exhaust lap, is increased an amount equal to $L S$, the release will take place later at the crank position $O B_2$, and compression will take place earlier at $O D_2$. The

contrary effect will be observed by decreasing the inside lap, or exhaust lap.

If the angular advance is increased, all the events will occur earlier, as is evident from Fig. 30. The crank revolves in the direction indicated by the arrow and $O\,A_2$ (new position of admission) is *ahead* of $O\,A_1$ the old position.

If the eccentricity is increased, Fig. 31, the valve travel will increase and admission will take place earlier at $O\,A_2$; the lead will be increased an amount equal to I_1I_2, and cut-off will take place later at $O\,C_2$. Release will be earlier at OB_2 and compression will be later at $O\,D_2$. The upper valve circle will now cut the arc drawn from O as a center, with a radius equal to the outside lap plus the width of steam port, in the points W_2 and H_2, and the admission port will be open wide while the crank is moving from $O\,W_2$ to $O\,H_2$. Similarly, the lower valve circle cuts the arc drawn from O as a center, with a radius equal to the inside lap plus the width of steam port, in the points W_1 and H_1. The steam port is then wide open to exhaust while the crank is moving from W_1 to H_1. From the above, it will be seen that the periods are all changed by changing the travel, thus admission and exhaust begin sooner and last longer, and expansion and compression begin later and cease sooner.

For convenience, these results are collected in Table I, which shows the effect of changing the laps, travel, and angular advance.

There are, of course, all sorts of combinations that would make up different problems, but they can all be solved in the same general way, as they are modifications of the problems solved above.

DESIGN OF SLIDE VALVE

In designing a slide valve, some of the variables are assumed and the others are found by means of the diagrams presented above. These diagrams show only the dimensions of the inside and outside laps and travel of valve; the other dimensions of the valve and seat must be calculated.

Area of Steam Port. *Steam Supply Pipe.* It is generally conceded by authorities that the pipes supplying steam to steam engines should be of such dimensions that the mean velocity of steam in them would not exceed 6,000 feet per minute. If the velocity of steam exceeds 6,000 feet per minute, there will be a very appreciable

loss of pressure, which is objectionable. In computing the size of a steam supply pipe for an engine, the assumption is made that the cylinder is filled at each stroke. The volume of steam passing through the steam pipe must equal the total volume of steam used by the cylinder.

Let d equal diameter of steam pipe in inches; D equal diameter of cylinder in inches; L equal length of stroke in feet; and N equal revolutions per minute (r. p. m.).

The area of the steam pipe in square feet would be $\frac{\pi d^2}{4\times 144}$ and that of the cylinder would be $\frac{\pi D^2}{4\times 144}$. The total volume of steam flowing through the pipe per minute would be $\frac{\pi d^2}{4\times 144}\times 6000$. Disregarding the volume of the piston rod, the total volume of steam used by the cylinder in one minute would be $\frac{\pi D^2}{4\times 144}\times 2LN$.

Since the volume of steam flowing through the pipe per minute must equal that used by the cylinder in the same time, we can equate the two expressions; that is,

$$\frac{\pi d^2}{4\times 144}\times 6000=\frac{\pi D^2}{4\times 144}\times 2LN$$

Solving,

$$d^2=\frac{D^2LN}{3000}$$

$$d=\frac{D\sqrt{LN}}{54.772}$$

Exhaust Pipe. For exhaust pipes, the mean velocity of steam is taken as 4,000 feet per minute. Therefore

$$\frac{\pi d^2}{4\times 144}\times 4000=\frac{\pi D^2}{4\times 144}\times 2LN$$

Solving,

$$d^2=\frac{D^2LN}{2000}$$

$$d=\frac{D\sqrt{LN}}{44.721}$$

EXAMPLE. Suppose an engine is 10 inches × 18 inches, and makes 180 revolutions per minute. Determine the diameters of the steam and exhaust pipes.

SOLUTION. Substituting in the equation

$$d = \frac{D\sqrt{LN}}{54.772}$$

gives for the diameter of the steam supply pipe

$$d = \frac{10\sqrt{1.5 \times 180}}{54.772}$$

$$= \frac{164.3}{54.772}$$

$$= 3 \text{ inches}$$

The required diameter of exhaust pipe would be

$$d = \frac{D\sqrt{LN}}{44.721}$$

$$= \frac{10\sqrt{1.5 \times 180}}{44.721}$$

$$= \frac{164.3}{44.721}$$

$$= 3.67 \text{ inches}$$

A 4-inch pipe would probably be used.

In practice different builders use different formulas, but all are derived from the fundamental assumptions made above, with certain constants added for different types of engines. The size of both steam and exhaust pipes required for engines of the same class is not affected in any marked degree by different types of valve gears.

For a very large engine cutting off early, the allowable velocity may be taken as 8,000 feet per minute instead of 6,000 feet.

Width of Steam Port. The port opening at admission should give nearly as great an area as the steam pipe in order to prevent loss of pressure due to wire-drawing, but the actual width of the port should be great enough for the free exhaust of steam. It is well to have the steam port a little larger than the area of the steam pipe, then with a port opening of six-tenths to nine-tenths of the port area for admission and full port opening at exhaust, satisfactory conditions will result.

The length of the ports is usually made about eight-tenths the diameter of the cylinder. Then in the 10-inch $\times$ 18-inch engine, the steam ports would be $.8 \times 10$, or 8 inches long. If the area for admitting steam is 7.0686 square inches (corresponding to a pipe 3 inches in diameter) and the length of port is 8 inches, the width will be $\frac{7.0686}{8}$, or .8836 inch—about $\frac{7}{8}$ inch.

The width of port opening would be about $.9 \times .8836$, or .79524 inch—about $\frac{13}{16}$ inch.

Width of Exhaust Port. When the slide valve is at its maximum displacement, the valve overlapping the exhaust port, as shown in Fig. 7, reduces the area more or less. In designing the valve, the exhaust port should be of such a width that the maximum displacement of the valve does not reduce the area of the exhaust port to less than the area of the steam port. It is not advisable to make the exhaust port too large, for this increases the size of the valve and thus causes excessive friction.

The height of the exhaust cavity should never be less than the width of the steam port and may be made much higher to advantage.

Width of Bridge. The bridge must be of sufficient width so that the outside edges of the valve can not uncover the exhaust port. The width of the steam port plus the width of the outside lap plus the width of the bridge must be greater than the maximum displacement.

The width of the bridges should not be less than the thickness of the cylinder wall in order to make a good casting.

Point of Cut-Off. In the study of Steam Engine Indicators, it was shown that if the point of cut-off is too early, the other events are not good. If a plain slide valve is used with an automatic cut-off, the point of cut-off is controlled either by changing the eccentricity or by changing the angular advance. Either of these methods will accomplish the result at the expense of the compression, which at a very early cut-off may be excessive. Except for locomotives and high-speed engines, where compression is an advantage, the plain slide valve is not arranged to cut off earlier than one-half or two-thirds stroke. If an earlier cut-off is desired, large outside laps are necessary.

Lead. The lead of stationary engines varies from zero to $\frac{3}{8}$ inch according to the style of engine and type of valve gear. An engine having high compression that compresses the steam nearly to boiler pressure will give good results with little or no lead. If the ports are small and the clearance large, there should be considerable lead in order to insure full initial pressure on the piston at the beginning of the stroke. Valves that open slowly require more lead than quick-acting valves.

ILLUSTRATIVE PROBLEM

EXAMPLE. Design and lay out the valve and valve seat for an engine of cylinder diameter 10 inches, stroke 18 inches, revolutions 180 per minute, lead angle 3 degrees, cut-off equal at both ends and taking place at 75 per cent of stroke, maximum port opening .9 area of steam pipe, compression 15 per cent of the stroke at both ends, and length of connecting rod 3 feet.

SOLUTION. The piston displacement, or cylinder volume, will be

$$\frac{3\ 1416\times 10^2}{4}\times 18 = 1413.7 \text{ cubic inches, or .818 cubic feet.}$$

If the engine makes 180 revolutions, neglecting the volume of the piston rod, it will use $2\times 180\times .818 = 294.48$ cubic feet of steam per minute. Steam pipe area $= \frac{294\ 48}{6000} = .0491$ square feet, or 7.07 square inches.

This 7.07 square inches would also be the least possible area of the steam ports. If the length of port is made eight-tenths the diameter of cylinder, the width will be $\frac{7\ 07}{8} = .88$ inch, or about $\frac{7}{8}$ inch. The width of maximum port opening will be $9\times 88 = .792$ inch, or nearly $\frac{13}{16}$ inch.

Zeuner Diagram. It will be necessary to draw a separate valve circle for each end of the cylinder. First, consider the head end. The valve travel not being known, we shall lay off $X\,Y$ on an assumption of 6 inches travel and draw the eccentric circle as shown in Fig. 32. Lay off the lead angle $X\,O\,A_1 =$ 3 degrees. Lay off $X\,C_2 = .75$ of the assumed valve travel $4\frac{1}{2}$ inches. Draw the arc C_1C_2, as previously explained, and draw $O\,C_1$ which will be the crank position at the point of cut-off. The radius of the arc C_1C_2 will be equal to four times the radius of the eccentric circle, or 12 inches, because the connecting rod is four times the length of the crank. Let the line $O\,E_1$ bisect the angle $A_1O\,C_1$, and on $O\,E_1$ draw the valve circle. $O\,V_1$ $(=O\,K_1)$ is then the outside lap, with these assumed conditions. Drawing the lap circle, the maximum port opening E_1N_1 is found to equal $1\frac{7}{16}$ inches, although $\frac{13}{16}$ is all that is necessary. The assumed eccentricity is 3 inches, therefore the probable eccentricity is found from the proportion

$$x : 3 :: \tfrac{13}{16} : 1\tfrac{7}{16}$$

$$x = 1\tfrac{11}{16} \text{ inches}$$

Now draw a new eccentric circle with a radius of $1\frac{11}{16}$ inches and a new valve circle with a diameter $O\,E_2$ $1\frac{11}{16}$ inches. $O\,K_2$ is now the outside lap and

the maximum port opening is equal to $E_2 N_2$, which from actual measurement is found to be $\frac{13}{16}$ inch. The outside lap $O\,K_2$ ($=O\,V_2$) is $\frac{27}{32}$ inch and the lead $I\,J$ is $\frac{1}{32}$ inch.

Produce E_1O to F and draw another valve circle. We shall use this valve circle to determine the outside laps and lead for the crank end of the cylinder. Since the cut-off is to be .75 of the stroke, we may lay off $O\,H_2 = O\,C_2$ and, with a radius of 12 inches, draw the arc H_1H_2. Then, as already explained, $O\,H_1$

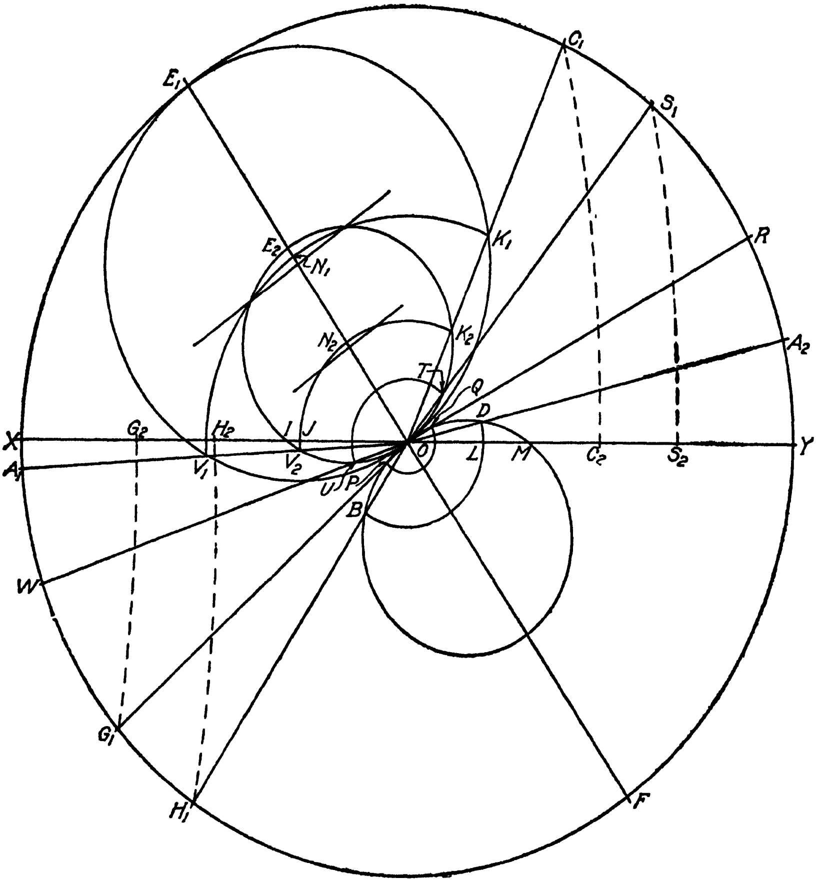


Fig 32. Zeuner Diagram for Design of Valve and Valve Seat in Problem Page 35

will be the crank angle at cut-off on the return stroke. $O\,B$, the outside lap, will be $\frac{19}{32}$ inch. Draw the lap circle intersecting the valve circle at D. Then $O\,D\,A_2$ is the crank position at admission on the return stroke and $L\,M$, $\frac{3}{8}$ inch is the lead on the crank end of the cylinder The maximum port opening will always be greater at the crank end than at the head end because the crank end lap is less in order to get the equal cut-off. If the laps were equal, of course the port openings would be equal.

Now lay off $X\,G_2$ equal to fifteen-hundredths of $X\,Y$ and find the crank position $O\,G_1$. This is the compression on the head end of the cylinder and gives an inside lap on this end of $\frac{7}{32}$ inch, which is equal to $O\,P$. Draw the lap circle $P\,Q$, which allows us to draw through Q the crank line $O\,R$, which is the release on the forward stroke.

Lay off $Y\,S_2$ ($=X\,G_2$) equal to fifteen-hundredths of $X\,Y$, and construct the crank line $O\,S_1$, which is the crank position at the crank-end compression $O\,S_1$ intersects the valve circle at T, giving $O\,T$, $\frac{7}{16}$ inch, as the inside lap on the crank end Draw this lap circle, which will intersect the valve circle at U This enables us to draw $O\,U\,W$, the crank position at release, on the return stroke.

Layout of Valve. From the data determined by means of these diagrams, the valve may now be laid out. For convenience let us tabulate the results obtained as follows:

DATA	HEAD END	CRANK END
Cut-off (per cent of stroke)	75 per cent	75 per cent
Outside lap	$\frac{27}{32}$ inch	$\frac{19}{32}$ inch
Inside lap	$\frac{7}{32}$ inch	$\frac{7}{16}$ inch
Lead	$\frac{3}{32}$ inch	$\frac{3}{8}$ inch
Port opening	$\frac{13}{16}$ inch	$1\frac{1}{16}$ inches
Width of port	$\frac{7}{8}$ inch	$\frac{7}{8}$ inch

Fig. 33 shows this valve in section. Let us begin at the end having the largest inside lap or, in this case, at the crank end. Lay out the steam port $\frac{7}{8}$ inch wide and the crank-end outside lap $\frac{19}{32}$ inch. The bridge will be, say, $\frac{3}{8}$ inch wide. From the inner edge of the steam port, lay off the crank-end inside lap $\frac{7}{16}$ inch. When the valve moves to the left, the point E_2 will travel $1\frac{11}{16}$ inches—a distance equal to the eccentricity—and in this position of extreme displacement, the exhaust port E_1F must be open an amount at least equal to the steam port, $\frac{7}{8}$ inch. Therefore, we lay off $E_1\,F$ equal to $1\frac{11}{16}$ inches + $\frac{7}{8}$ inch = $2\frac{9}{16}$ inches. The inside lap overlaps the bridge nearly $\frac{1}{8}$ inch, so that we shall have to make the exhaust port opening equal to $2\frac{5}{8}$ inches. Lay off $\frac{3}{8}$ inch again for the bridge and measure back $\frac{7}{32}$ inch, equal to the head-end inside lap. The port is $\frac{7}{8}$ inch wide, and the head-end outside lap of $\frac{27}{32}$ inch completes the outline of the valve seat.

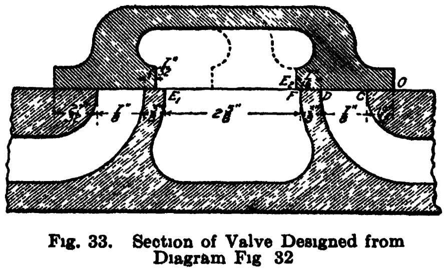


Fig. 33. Section of Valve Designed from Diagram Fig 32

Reversing Simple Engine. In the operation of a simple engine having a plain slide valve or a piston valve, it sometimes becomes necessary to reverse the direction of rotation of the engine shaft. Remembering the principles presented in the foregoing study of the Zeuner diagram, this is not a difficult task.

It is proposed to here show *first*, how an engine may be reversed with a direct valve, engine running over; *second*, with a direct valve, engine running under; *third*, with an indirect valve, engine running over; and *fourth*, with an indirect valve, engine running under.

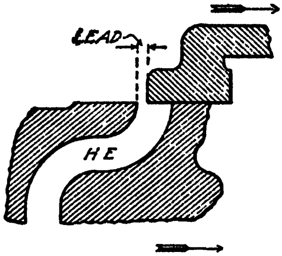


Fig. 34. Section Showing Lead of Valve, Engine Running Over

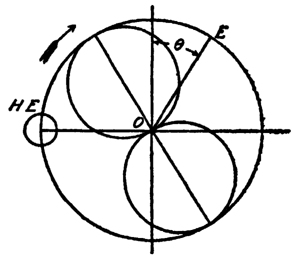


Fig. 35 Diagram for Direct Valve, Engine Running Over

Definitions. Before explaining the operation for obtaining the above, it is well to have an understanding of the meaning of the terms "direct" and "indirect" as applied to a valve, and of "running over" and "under" as applied to an engine.

A valve is said to be a *direct*, or *outside admission, valve*, when at the beginning of the stroke the valve and the piston are moving in the same direction, as indicated by the arrows in Fig. 6. It is

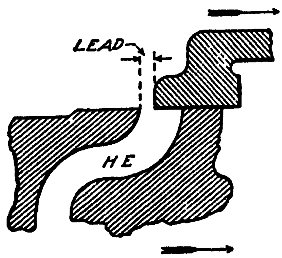


Fig 36 Section Showing Lead for Direct Valve, Engine Running Under

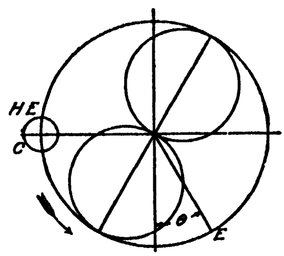


Fig 37. Diagram for Direct Valve, Engine Running Under

also to be noted that steam is being admitted to the cylinder by the outer edge of the valve, which is the reason for calling it an outside admission valve.

If, in Fig. 6, the valve should be moving in the opposite direction from that shown and steam should be entering the cylinder by the inner edge of the valve, the valve would then be said to be an *indirect*, or *inside admission, valve*.

Most plain slide valves are of the outside admission type, while most piston valves are of the inside admission type.

An engine is said to be *running over*, if, when the piston is moving from the head end toward the crank end, the moving parts, such as connecting rod, crank, etc., are above the center line, as shown in Fig. 17. The engine is said to be *running under* when the above mentioned parts are below the center line when the piston is moving from the head end toward the crank end.

Direct Valve, Engine Running Over. In Fig. 34, let the valve have lead equal to that shown. Since this is a direct valve, engine running over, the valve will be to the right of its mid-position and moving to the right, hence the eccentric will be $(90+\theta)$ degrees ahead of the crank. If the engine is on the head-end dead center, the eccentric would be at E, that is, $(90+\theta)$ degrees ahead of the crank. The right and left valve circles will be located in the second and fourth quadrants, respectively, as shown in Fig. 35.

Direct Valve, Engine Running Under. With a direct valve, engine having lead and running under, as illustrated in Figs. 36 and 37, the valve will be in the same relative position as in the former case, when the crank is on the head-end dead center. In this position the valve must be to the right of its mid-position and moving towards the right, hence the eccentric must be, as shown at E, Fig. 37, an angular distance of $(90+\theta)$ degrees ahead of the crank.

The right and left valve circles will be located in the first and third quadrants, respectively, as shown in Fig. 37.

It is to be noted on comparing the position of the eccentric in Figs. 35 and 37 that both of the eccentric positions make an angle equal to the angle of advance with the vertical. Therefore, to reverse a direct valve, engine running over, turn the eccentric around the shaft, in the direction in which the engine is running, by an angle of $(180-2\theta)$ degrees, or turn the eccentric ahead of the crank, in the direction in which the engine is to run, an angle of $(90+\theta)$ degrees.

Indirect Valve, Engine Running Over. An indirect valve engine running over is illustrated in Figs. 38 and 39. Remembering that the valve must be moving to the left as the piston moves from the head end toward the crank end, and that the valve must be displaced by an amount equal to the lap plus the lead to the left of its mid-position, the eccentric must be below the horizontal and behind

the crank an angular distance of $(90-\theta)$ degrees. Hence, it is located at E, Fig. 39. The right and left valve circles will be located in the fourth and second quadrants, respectively

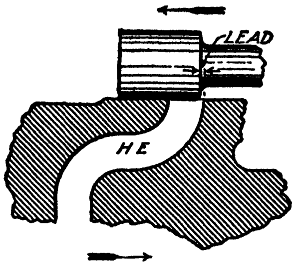


Fig 38. Section of Indirect Valve, Engine Running Over

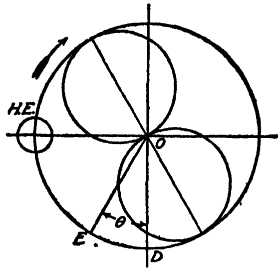


Fig 39 Diagram of Indirect Valve, Engine Running Over

Indirect Valve, Engine Running Under. To locate the eccentric for an indirect valve engine having lead and running under (see Figs. 40 and 41), proceed as before. The eccentric will be found at E, Fig. 41, and the right and left valve circles will be located in the first and third quadrants, respectively.

An examination of Figs. 38 to 41 will disclose the fact that to reverse an engine using an indirect valve, it is only necessary to turn the eccentric through an angle of $(180-2\theta)$ degrees in the direction in which the engine shaft is turning or, in other words, the procedure is the same as for a direct valve.

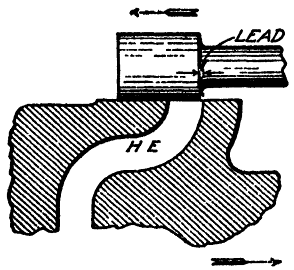


Fig 40 Section of Indirect Valve, Engine Running Under

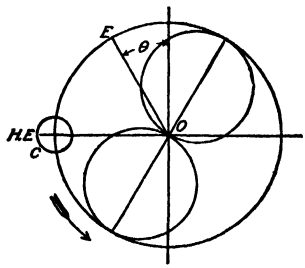


Fig 41 Diagram for Indirect Valve, Engine Running Under

Comparisons and Comments. A comparison of Figs. 34 and 35 with 38 and 39 will indicate the relative positions of the eccentric for an engine running over with a direct valve and for one running over with an indirect valve. It is evident that in the first case, the eccentric precedes the crank by an angle of $(90+\theta)$ degrees, whereas

in the second, the eccentric follows the crank by an angle of $(90-\theta)$ degrees. This same condition is true for two engines running under, one using a direct valve and the other an indirect.

As an aid in locating the valve travel circles after the eccentric position has been determined, remember that the quadrant separated by a vertical line through the center, from the quadrant containing the eccentric position, is the quadrant in which the right valve travel circle is to be located.

All of the study on the Zeuner valve diagram thus far has to do with an engine running over having a direct valve. After the location of the eccentric position has been determined for the above various conditions, the construction of the Zeuner diagram should be a simple matter.

The principles underlying the location of the eccentric for an engine running over or under and having a direct or indirect valve should be borne in mind when setting valves.

VALVE SETTING

Possible Adjustments. The principles of valve diagrams are useful in setting valves as well as in designing them. The valve is usually set as accurately as possible, and then, after indicator cards have been taken, the final adjustment can be made to correct slight irregularities.

The slide valve is so designed that the laps can not be altered without considerable labor, and the throw or eccentricity of the eccentric, which determines the travel of the valve, is usually fixed. The adjustable parts are commonly the length of the valve spindle and the angular advance of the eccentric.

By lengthening or shortening the valve spindle, the valve is made to travel an equal distance each side of the mid-position. Moving the eccentric on the shaft makes the action of the valve earlier or later as the angular advance is increased or decreased.

To Put Engine on Center. It is usual to put the engine on center before setting the valve. First, put the engine in a position where the piston has nearly completed the outward stroke and make a mark M_1, Fig. 42, on the guide opposite the corner of the crosshead at some convenient place. Also make a mark P with a center punch on the frame of the engine near the crank disk. With

this mark P as a center, describe an arc C on the wheel rim with a tram.*

Turn the engine past the center until the mark on the guide again corresponds with the corner of the crosshead and make another

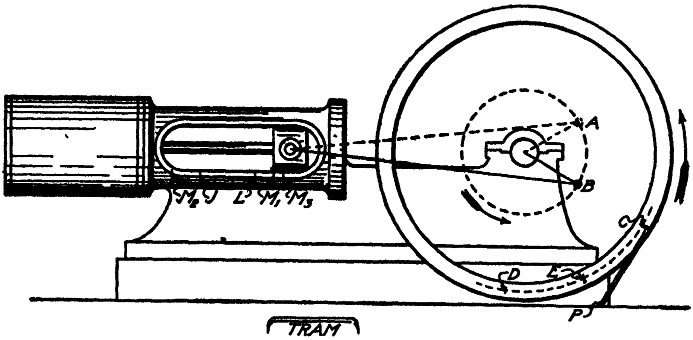


Fig. 42. Sketch of Engine, Showing Method of Putting Engine on Center

mark D on the wheel with the tram, keeping the same center. With the center of the pulley, or crank disk, as a center, describe an arc CD on the rim, which intersects the two arcs drawn with the tram. Bisect the arc CD and turn the engine until the new point is distant from the point P an amount equal to the length of the tram, in which position the engine will be on center.

The engine should always be moved in the direction in which it is to run so that the lost motion of the wrist pin and crank pin

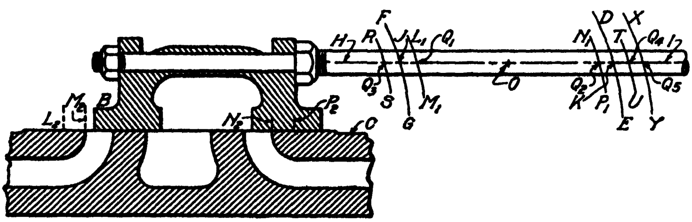


Fig. 43. Diagram Showing How Valve is Set for Equal Lead

will be taken up the right way. In case the engine has been moved too far at any time, it should be turned back beyond the desired point while the engine is moving in the proper direction. In this manner, the dead center can be located for both the head and crank ends.

*A tram is a steel rod with its ends bent at right angles and sharpened.

To Set Valve for Equal Lead. After locating the dead center points as described above the next step is to locate what are known as the port marks. In Fig. 43 move the valve to the left until cut-off occurs on the head end or until the edge of the valve at B is at M_2. Then, with a center C on some fixed point on the cylinder or engine frame, describe with a tram the arc $F\ G$ on the valve rod. Continue the rotation of the engine in the same direction until cut-off takes place at the crank end. Then with the same tram and center C, sweep the arc $D\ E$ on the valve rod. Draw the center line $H\ I$ and where this center line cuts the arcs $F\ G$ and $D\ E$, mark the points J and K, respectively, which points are known as the port marks. Bisect the distance between J and K, thus establishing the point O. When one tram point is in C and the other just enters the point J, the valve is just cutting off on the head end; and when the tram point coincides with C and K, it is an indication that cut-off is occurring on the crank end, hence a basis of comparison has been established for the two ends. Place the engine on the forward dead center and sweep the arc L_1M_1. The distance between the arcs L_1M_1 and $F\ G$, which is equal to JQ_1, represents the amount the valve extends over the port when the engine is on the head-end dead center. In a like manner, establish the arc N_1P_1 when the engine is on the crank-end dead center, in which position the valve overlaps the steam port the distance KQ_2. In order to have equal travel of the valve on either side of its mid-position, the distance $J\,Q_1$ should equal $K\,Q_2$. If necessary to equalize these distances, lengthen or shorten the valve stem as required. Having secured an equal valve travel, place the engine on the forward dead center. Since the engine is running under (see Fig. 37), the eccentric will be found $(90 + \theta)$ degrees ahead of the crank in the direction the engine is to run. Lay off on the valve stem the distances $J\,Q_3$ and $K\,Q_4$ equal to the required lead. With the tram point in C and the engine placed on the head-end dead center, turn the eccentric in the direction in which the engine is to run—which is *under* in this case—until the arc $R\ S$ passes through the point Q_3. Fasten the eccentric and turn the engine around until it is on the crank-end dead center. Sweep another arc as $T\ U$ with the tram. If this arc passes through the point Q_4, then the valve is correctly set for equal lead, that is, JQ_3 is equal to KQ_4. If, however, the arc $T\ U$

does not pass through the required point Q_4, but falls beyond, it is an indication of unequal lead, so a correction must be made. Suppose, for instance, that when the crank was placed on the crank-end dead center, the arc described from C fell at XY instead of TU, then it is obvious that the crank end has more lead than the head end. To make a correction for this inequality, find the difference between the lead on the head and crank ends—which in this case is equal to the distance Q_4Q_5—and correct half of the difference on the valve stem and the other half by altering the angle of advance. In this case, the valve stem should be lengthened by the amount $\frac{Q_4Q_5}{2}$, which would increase the lead on the head end by that amount and decrease it by the same amount on the crank end. After establishing an equal travel of the valve by adjusting the length of the valve stem, thus giving an equal amount of lead at each end, the desired amount of lead may be obtained by changing the angle of advance. To obtain the required lead in this case, it would be necessary to reduce the angle of advance. It may be necessary to make several trials before the desired results are obtained, this being particularly true if working on an engine having lost motion in the various parts. In order to eliminate the effect of lost motion in so far as possible, the engine should always be turned in the direction which it is to run.

In case it is difficult to turn an engine, the following method may be used. First, loosen the eccentric on the shaft and turn it around until it gives a maximum port opening first at one end and then at the other. If the maximum port openings are not equal, make them so by changing the length of the valve spindle by half the difference. When the above adjustment has been made, set the engine on dead center and give the valve the proper lead by turning the eccentric on the shaft. The angular advance is thus adjusted.

To Set Valve for Equal Cut-Off. To set the valve for equal cut-off, observe the preliminary steps of locating on the valve stem the dead-center points, port marks, and equal travel of the valve to either side of its mid-position, as described in connection with setting the valves for equal lead.

Assume that it is desired to set the valves for an equal cut-off of 75 per cent. On the guides of the engine illustrated in Fig. 42,

locate the points M_2 and M_3, corresponding to the extreme positions of the edge of the crosshead, or a given point on the crosshead. The distance M_2M_3 represents the stroke of the piston, so when 75 per cent cut-off occurs, the reference point on the crosshead should be at a point J, which is 75 per cent of the stroke M_2M_3 for the crank end and at the point L for 75 per cent cut-off on the head end. Remembering that the points J and K on the valve stem in Fig. 43 represent points of cut-off, all required reference points needed are known. Turn the engine over in the direction indicated in Fig. 42 until the reference point on the crosshead corresponds to the reference point on the guide, as L, for the head-end cut-off. Then with the tram in the center C, Fig. 43, describe an arc, say, L_1M_1. Continue the rotation of the engine in the same direction until the piston has completed the forward stroke and has returned to the point where the reference lines on the crosshead and the guide J coincide. Tram the valve stem as before, locating the arc, say, N_1P_1. Since the tram should coincide with the arcs $F\,G$ and $D\,E$ for the head-end and crank-end cut-off, respectively, it is therefore evident that with the tram coinciding with L_1M_1 and N_1P_1 that the required cut-off is not obtained but occurs too early. Since the distances Q_1J and Q_2K are equal, the length of the valve stem does not need to be disturbed. To make cut-off occur later, decrease the angle of advance by moving the eccentric opposite to the direction in which the engine is to run. For instance, with the engine standing so that the point L, Fig. 42, and the end of the crosshead are coincident, move the eccentric until the tram points coincide with C and J, Fig. 43. Try the points for the cut-off on the crank end, and if the tram fits easily into C and K, then the valve is set correctly. If, however, the tram points do not fit into the points C and K, continue the operation until the desired results are obtained.

From the above discussion, two points have been established:

(1) Moving the valve on the valve rod changes the corresponding events the same on both ends, one being made earlier and the other later. That is, if the cut-off is made earlier on the head end, it will be later on the crank end, and so on for the other events.

(2) Moving the eccentric on the shaft or changing the angle of advance changes the corresponding events the same for both ends, both being made earlier or both later.

MODIFICATIONS OF THE SLIDE VALVE

Balancing Steam Pressure. The ordinary slide valve is most suitable for small engines. For engines of large size, some method must be employed to balance the steam pressure on the back of the valve. With large valves, such for instance as those of locomotives or large marine engines, a great force is exerted by the steam, and the valve is forced against its seat so hard that a large amount of

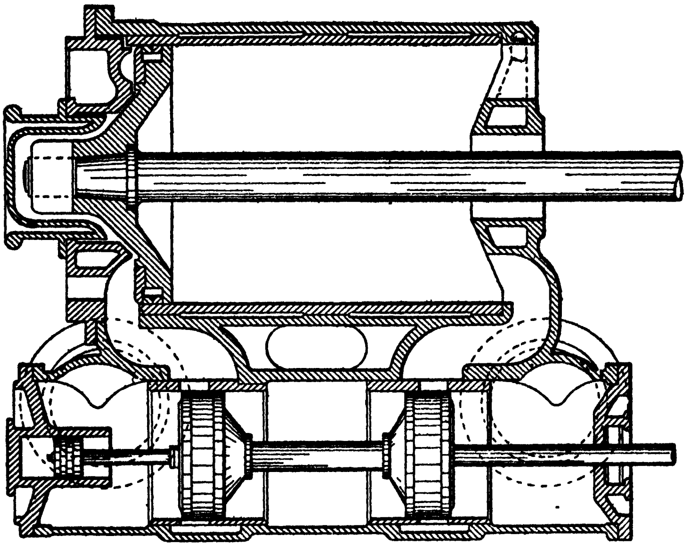


Fig 44 Section of Piston Valve and High-Pressure Cylinder of U. S S. "Massachusetts" Showing Method of Balancing

power is necessary to move it. This excessive pressure causes the valve to wear badly and is a dead loss to the engine. The larger the valve, the greater this loss will be.

Piston Valve. To prevent excessive pressure on the back of the valve, the piston valve is commonly used, especially in marine engines. This valve consists of two pistons which cover and uncover the ports in precisely the same manner as the laps of the plain slide valve. These pistons are secured to the valve stem in an approved manner and are fitted with packing rings.

The valve seat consists of two short cylinders or tubes accurately bored to fit the pistons of the valve. The port openings are not continuous as in the plain slide valve, but consist of many small openings, the bars of metal between these openings preventing the packing rings from springing out into the ports.

Steam may be admitted to the middle of the steam chest and exhausted from the ends or *vice versa.* With the former method, the live steam is well separated from the exhaust, and the valve-rod stuffing box is exposed to exhaust steam only. This is a good arrangement for the high-pressure cylinder; if used for a cylinder in which there is a vacuum, air may leak into the exhaust space through the valve-rod stuffing box. With this arrangement, the steam laps must be inside and the exhaust laps on the outside ends.

The piston valve may be laid out and designed by means of the Zeuner diagram just as if it were a plain slide valve, and the action

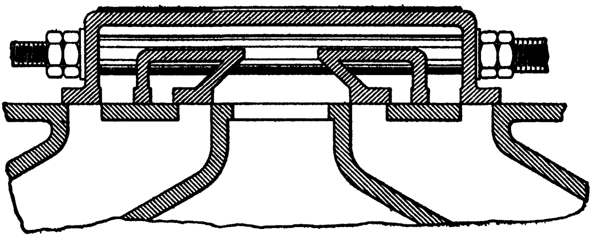


Fig. 45. Section of Double-Ported Slide Valve

is the same except that it is balanced so far as the steam pressure is concerned, the power to drive it being only that necessary to overcome the friction due to the spring rings.

Fig. 44 shows a section of the piston valve and the high-pressure cylinder for one of the engines of the U. S. S. "Massachusetts." This valve consists of two pistons connected by a sleeve through which the valve rod passes. This valve rod is prolonged to a small balancing piston, placed directly over the main valve. The upper end of the balancing cylinder does not admit steam, so that the steam pressure below the balancing piston will practically carry the weight of the piston valve, thus relieving the valve gear and making the balance more nearly complete.

Double-Ported Valve. Sometimes it is impossible to get sufficient port opening for engines of large diameter and short stroke,

especially those having a plain slide valve with short travel. This difficulty may be overcome by means of the double-ported valve shown in Fig. 45. It is equivalent to two plain slide valves, each having its laps. The inner valve is similar to a plain slide valve except there is communication between its exhaust space and the exhaust space of the outer valve. Each passage to the cylinder has two ports; a bridge separates the exhaust of the outer valve from the steam space of the inner valve, and the outer valve is made long enough to admit steam to the inner valve.

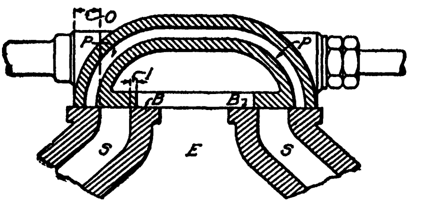


Fig. 46. Trick Valve Shown in Mid-Position

This valve may be considered as equivalent to two equal slide valves of the same travel, each having one-half the total port opening. To admit the same amount of steam as a plain slide valve, the double-ported valve requires but half the valve travel; this is advantageous in high-speed engines.

To balance the excessive steam pressure, the back of the valve is sometimes provided with a projecting ring which is fitted to a similar ring within the top of the valve chest. These rings are planed true and fit so that steam is prevented from acting on the back of the valve.

Trick Valve. The defect of the plain slide valve, due to the slowness in opening and closing, is largely remedied in the trick valve,

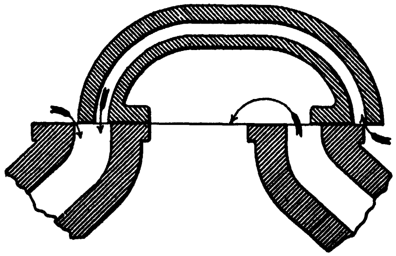


Fig. 47 Trick Valve Showing Admission of Steam Just Beginning

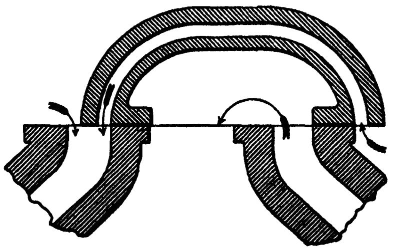


Fig 48 Trick Valve at Extreme Right Position with Steam Port Open Wide

which is so made that a double volume of steam enters during admission. Thus a quick and full opening of the port is obtained with a small valve travel.

In Fig. 46 the valve is shown in mid-position. It is similar to a plain slide valve except that there is a passage *P P* through it. It has an outside lap *O* and an inside lap *I*. The seat is raised and has steam ports *S S*, bridges *B B*, and exhaust port *E*. If the valve moves to the right a distance equal to the outside lap plus the lead, it will be in the position shown in Fig. 47. Steam will be admitted at the extreme left edge of the valve just the same as though it were a plain slide valve; also, since steam surrounds the valve, it will be admitted through the passage as shown in Fig. 47. If the lead is the same as for a plain slide valve, $\frac{1}{16}$ inch for instance, this valve would give double the port opening, that is, $\frac{1}{8}$ inch, when the valve was open a distance equal to the lead.

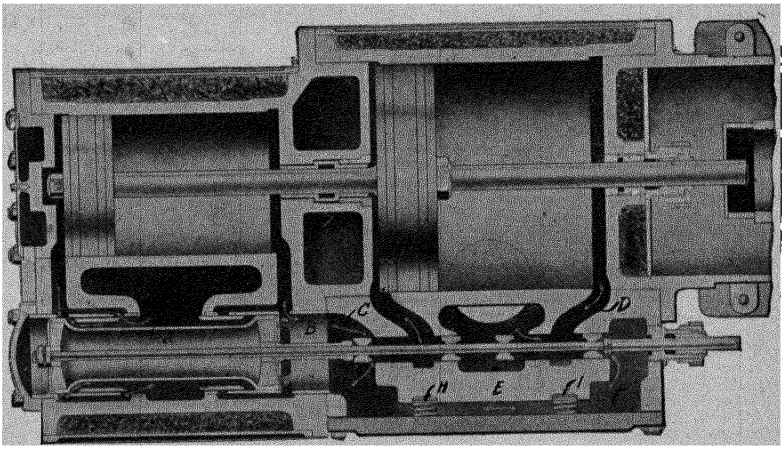


Fig 49 Obtaining Perfect Balance by Use of Double-Ported Piston Valve and Double-Ported Slide Valve in Compound Engine

Fig. 48 shows the valve when it is in extreme position to the right and the port is full open to steam.

Piston valves are also made with a passage similar to that of the trick valve for double admission, that used with the Armington and Sims engine being, perhaps, the best example.

Application of Various Types. Piston valves are commonly used on the high and intermediate cylinders of triple-expansion engines, and if well made and fitted with spring rings, should not leak. Small piston valves are often made without packing rings; but even if they fit accurately when new, they soon become worn and cause trouble.

The double-ported valve, the trick valve, and others, often have some device for relieving the pressure, such as a bronze ring or cylinder fastened to the back of the valve. This ring is pressed by springs against a finished surface of the valve chest cover, and the space thus enclosed by the ring may be connected to the exhaust. There are numerous devices for balancing valves, but they are usually more or less expensive and are liable to cause trouble from leakage.

Fig. 49 well illustrates the application of a double-ported piston valve and a double-ported slide valve to a compound engine. It also shows a method used for obtaining a perfect balance. The piston valve on this engine is a hollow inside admission valve. The steam passes from the cavity *A* through the double ports in the piston valve, forcing the high pressure piston to the right, which action causes the exhaust steam to pass out of the high pressure cylinder through the passage *B* into the steam chest of the low pressure cylinder. The steam passes around the flat valve at *C C* into the low pressure cylinder. The steam back of the low pressure piston passes through the port *D* into the exhaust cavity. The pressure plate *E* is held against the flat slide valve by the springs *H* and *I*, there being steam all around the pressure plate, as at *F* and *G*. The valve fits closely between the valve seat and pressure plate, but the pressure plate being supported at the sides eliminates the pressure between the valve and its seat. Both of these valves are said by the builders to be in perfect balance.

Reversing Mechanism. In the early development of valve gears, it became necessary to devise some means of reversing the engine, hence it is found that a great many of the most prominent gears, such as the Stephenson, Walschaert, Marshall, and many others of more or less merit, embody the reversing feature.

Reversing by Means of One Eccentric. At first, the reversing of an engine was accomplished by the use of one eccentric, there being two methods by which this was done.

(1) The device shown in Fig. 50 was used on some of the earliest locomotives and marine engines, and may now be found as the reversing medium for engines used on small launches. The eccentric *E* is loose on the shaft between a fixed collar *G* and a hand wheel *H*. A stud projecting from the eccentric passes through a curved

slot in the disk of the wheel and can be clamped by a hand nut *F*. When running forward with the crank at *C*, the center of the eccentric is at *A* and the nut is clamped at *F*. To reverse, steam is shut off and, when the engine stops, the nut *F* is loosened and then moved to *B* and clamped; or, after *F* is loosened, the wheel, shaft, crank, and propeller are turned over by hand until *B* strikes the stud at *F*, where it is clamped. The engine will then run astern.

(2) The eccentric was mounted on a sleeve, which could be moved longitudinally along the shaft of the engine by means of a lever The sleeve had a spiral slot cut on the inside of it, which subtended an angle of $(180-2\theta)$ degrees. This slot fitted over a radial pin on the shaft, so when the sleeve was pushed in or out by the lever, both the sleeve and the eccentric were turned through $(180-2\theta)$ degrees, thus reversing the engine.

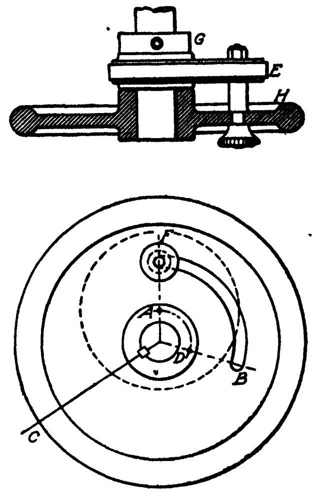


Fig 50 Early Reversing Device by Means of One Eccentric

Reversing by Means of Two Eccentrics and Gab-Hooks. It is obvious from the foregoing that the method of reversing by shifting one eccentric is awkward and not well adapted to high speeds and large engines. It was a natural transition, therefore, from the one eccentric to the more convenient reversing gears having two eccentrics, one set $(90+\theta)$ degrees ahead of the crank for the forward motion and one $(90+\theta)$ degrees behind the crank for the backward motion. At first, this arrangement was rather crude and objectionable in some respects, as will be noted later. The essential feature to be borne in mind with reference to a two-eccentric gear is, that the object is to have only one eccentric at a time operating the valve. In the early development, this was accomplished by using gabs or gab-hooks, which could be brought in contact with the valve rod at the pleasure of the operator. For instance, if the engineer wished to go forward, he would lower the arm *R*, Fig. 51, thus bringing B_1 in contact with the valve rod at *V*. The valve would then be operated by the forward eccentric E_1 and the engine would run

in the direction indicated by the arrow at E_1. To reverse the engine, B_1 would be disengaged and B_2 placed in connection with V. The valve would then be operated by the eccentric E_2, and the engine would run in the direction indicated by the arrow at E_2, which is the reverse of that indicated at E_1. It is to be particularly noted that only one eccentric actuates the valve at one time. All reversing gears of the two-eccentric type carry out this principle to a greater or less extent.

It will be obvious that the gab-hooks are an improvement over the shifting eccentric, but even they have certain objectionable features, the three principal ones being (1) the engine must have a slow speed of rotation; (2) the engine must be of such construction that

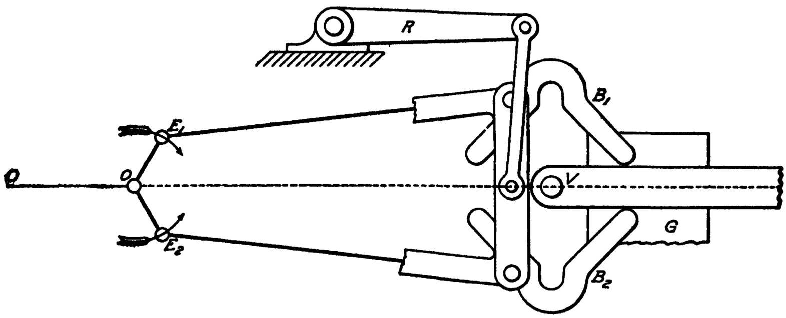


Fig. 51. Reversing Device Using Two Eccentrics and Gab-Hooks

the reversal can be accomplished in a leisurely manner—it is not convenient to reverse at a high speed with a gab-hook, but the engine must be turning slowly when the hook is dropped upon the pin; (3) the engine must be of such a type that it can be started by hand-working of the valves.

Reversing by Means of Two Eccentrics and Curved or Straight Links. To overcome these objectionable features, a step forward was taken when the gab-hooks were replaced by the curved or straight link, which is now used in connection with almost all reversing gears. This was a decided improvement as it not only accomplished the reversing of the engine but also made possible a variation in the adjustment of the valve mechanism, which permitted much more economical distribution of steam in the cylinder. There are two general classes of valve gears that use the curved link and its neces-

sary attachments, namely, the shifting link or the stationary link type, and the radial gear type.

The Stephenson gear is a worthy exponent of the shifting link type. The Walschaert, Joy, Marshall, and others are representatives of the radial gear type.

SHIFTING LINK TYPE OF VALVE GEAR

Stephenson Link Motion. As the Stephenson gear is one of the oldest reversing gears used and is perhaps the best known, a discussion of its principal features is presented first. This gear has been successfully used on stationary, traction, and marine engines, but its largest and, perhaps, most successful application has been on American locomotives. This gear is illustrated in Fig. 52. The two eccentrics E_1 and E_2, whose centers are at C_1 and C_2, respectively, are shown in their relative positions when the crank OA is at the crank-end dead center. The eccentric rods R_1 and R_2 are connected by forked ends to the link pins H and G. The link consists of two curved bars bolted together in such a manner that they may slide by the link block N. On the link are three sets of trunnions; the two outer ones, or link pins, are fitted into the forked end of the eccentric rods, and the middle one, known as the saddle pin, is fitted into the end of the drag links $F\,M$.

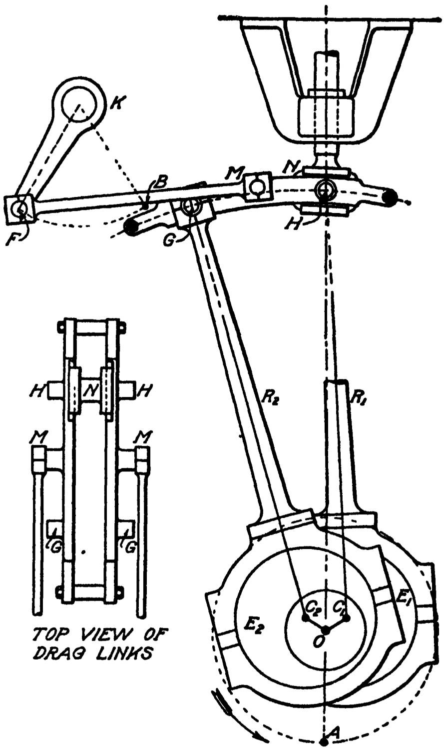


Fig 52. Stephenson Link Motion

The valve stem has, at its lower end, a pivoted block N, called the *link block*, provided with slotted sides through which the links

can slide, The reverse shaft, or rock shaft, K, here shown in the full gear "forward," may be turned until F moves to B; in this position the link will be pushed across the link block, and the valve will get its motion from the rod R_2 instead of from R_1, as before. The link in this position would be in full gear "backward."

From the foregoing, it is obvious that the Stephenson gear may be divided into three distinct mechanisms, each of which perform a definite function. *First*, the link motion proper, comprising the parts from the axle to the link; *second*, the adjusting gear, which is composed of the lifting shaft and reversing lever by which the power of the engine is controlled by lowering or raising the link; and, *third*, the valve and its attachments. The link motion proper is,

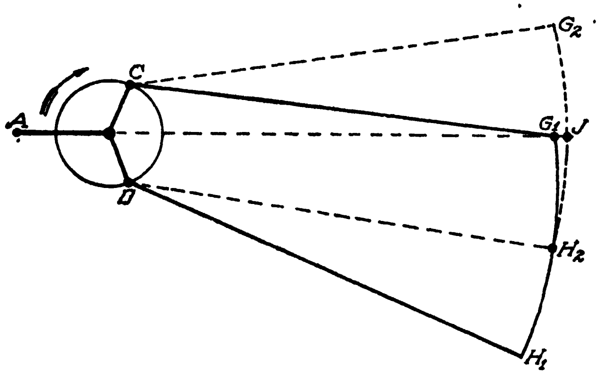


Fig. 53. "Open Rod" Arrangement of Eccentric Rods in Stephenson Gear

perhaps, the most important of the three, at least for the present study. Remembering that the link supplanted the gab-hook, it should be obvious that the eccentric rods and their connection to the link form a combination similar in action to the gab-hooks and valve rod, with some intervening parts which do not materially affect or change the operation.

Relative Position of Eccentric Rods. In order to have clearly in mind just what action does take place when the link is shifted from one position to another, it is essential that the relative position of the eccentric rods be understood. They are designated as "open rods" when arranged as shown in Fig. 53, with the eccentric centers C and D on the same side of the axle as the link, and "crossec rods" when the rods cross as illustrated in Fig. 54. The location length, and attachment of the eccentric rods to the link have a mate-

rial effect upon the movement of the valve. Experience and calculations have shown that the eccentric rods should not be shorter than eight times the throw of the eccentric. They are usually much longer than this. The distance between the eccentric rod pins should not be less than two and one-half times the throw of the eccentric. If the distance is less than this amount, the angle between the link and the block will be such that there will be an excessive slip of the block and undue stresses in the mechanism will be induced. The angularity of the eccentric rods produces irregularities in the movement of the valve, which can be largely compensated for by locating the saddle pin inside the center line of the arc, but not too far inside for then it would give a long slip of the link and be objectionable. The adjustment of the link also requires that special attention

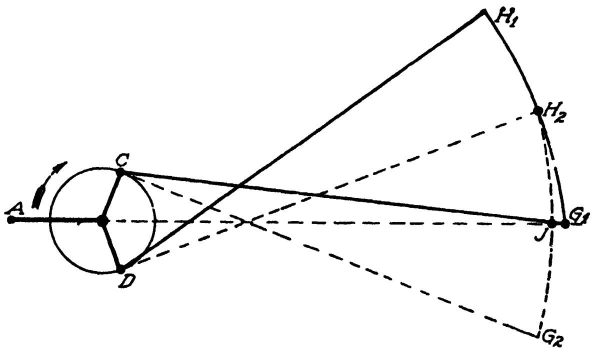


Fig 54. "Crossed Rod" Arrangement of Eccentric Rods in Stephenson Gear

tion be given to the amount of lead at full gear as well as to the increase of lead produced by "hooking up" the engine. With "open rods" it will be seen that when in full gear the link block is at G_1, and that if, without turning the crank, the link is shifted to mid gear, then the link block moves to J, Fig. 53, and the valve must consequently be moved toward the right an amount equal to G_1J, thereby increasing the lead on the crank end of the cylinder. With "crossed rods," moving the link from full to mid gear moves the link block from G_1 to J, Fig. 54, thus reducing the lead. It follows then that open rods give increasing lead from full toward mid gear, and that crossed rods give decreasing lead. With crossed rods there will be no lead when in mid gear. It will be apparent that the shorter the rods, the greater this increase or decrease will be.

The open rods are more generally used than the crossed rods; especially is this true in locomotive service. The feature of increasing lead from full to mid gear is the distinguishing characteristic of the Stephenson gear. When the engine is in full gear, so that the forward link pin G_1 is on the center line as in Fig. 53, then only the eccentric C controls the valve, and the travel of the valve will be equal to twice the throw of the eccentric C. In other words, when in full gear, only one eccentric moves the valve, as was the case when using gab-hooks. As the link is raised, both of the eccentrics have an effect on the motion of the valve, the result being very much the same as if another controlling eccentric of shorter throw were introduced. The throw of this resultant eccentric would decrease until the center was reached, when it would be a minimum. Finally, the

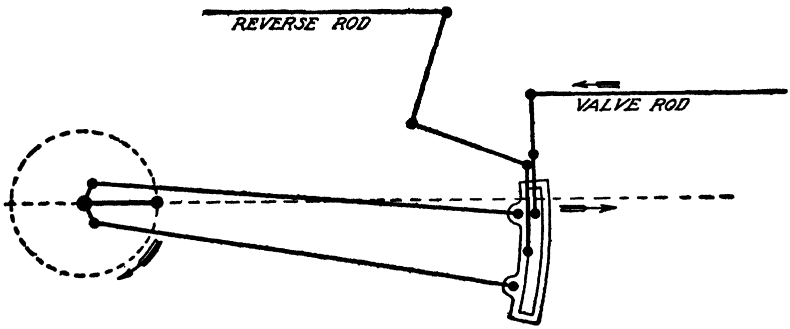


Fig. 55. Diagram of Stephenson Gear Showing Link Block and Rocker

center of this resultant eccentric would be on the center line of the motion, midway between the two actual eccentrics. At this point, the radius of the resultant eccentric would be equal to the sum of the lap and lead in full gear. Therefore, in mid gear, the valve travel is equal to twice the sum of the lap and lead in mid gear.

Location of Link Block. Nearly all marine engines, and some English locomotives, have their link blocks carried directly on the valve rod. American locomotives commonly use a rocker, one end of which carries the link block while the other moves the valve rod. This arrangement, indicated in Fig. 55, makes it possible to place the valve and steam chest above the cylinder. The position of the crank for the same valve position is just opposite that shown in Fig. 53, because the rocker reverses the direction of motion of

the valve. While apparently the crossed rod arrangement is used, yet it is really the open rod arrangement and gives increasing lead toward mid gear. A rod from the bell crank lever on the reverse shaft E leads back to the engineer's cab and connects with the reverse lever. This lever moves over a notched arc and may be held by a latch in any one of the notches, thus setting the link in any position from mid gear to full gear, either forward or back.

The Stephenson link is designed to give equal lead at both ends of the cylinder; but to accomplish this, the radius of the link arc (that is, an imaginary line in the center of the slot) must be equal to the distance from the center of this slot to the center of the eccentric. In Fig. 52, the radius of the link arc is equal to C_1H and C_2G.

Exact equality of lead is not essential, and the radius of the link arc is sometimes made greater or less than stated above in order to aid in equalizing the cut-off; but the change should never be great enough to affect the leads.

Application to Expansion and Cut-Off. Stephenson originally intended to use the link simply as a reversing gear, but soon found, however, that at intermediate points between the two positions of full gear, it would serve very well as a means of varying the expansion and cut-off. Very soon, the link came to be used not only on locomotives and marine engines, but on stationary engines as well, in connection with the reverse shaft which was under the control of the governor. The mechanism proved to be too heavy to be easily moved by a governor and it has gradually fallen into disuse on stationary engines excepting as a means of reversing.

In marine practice, the variable expansion feature is of little value, for marine engines run under a steady load and the link is set either at full gear or at some fixed cut-off. For locomotives, however, the variable expansion is nearly as important as reversing. Locomotives are generally started at full gear, admitting steam for nearly the entire stroke and then exhausting it at relatively high pressure. This wasteful use of steam is necessary to furnish the power needed in starting a train. After the train is under way, less power is required per stroke and the link is gradually moved toward mid gear or "hooked up" by the engineer, thus hastening the cut-off; the expansion is increased and the power is reduced in proportion to the load.

As the cut-off is changed, it is desirable to maintain an approximately equal cut-off at each end of the cylinder; this can be secured in the Stephenson gear by properly locating the saddle pin and the reverse shaft. When used without a rocker, as in Fig. 52, the saddle pin should be on the arc of the link or slightly ahead of it. When used with a rocker, the saddle pin should be behind the link arcs and, in order to give symmetrical action for both forward and backward running, it should be opposite the middle of the arc, that is, equally distant from each link pin.

Zeuner Diagram for Stephenson Gear. The Stephenson link can not be designed directly from the Zeuner diagram, but a systematic investigation can be made by using a wooden model of the proposed link. This can be mounted on a drawing board, and the effect of changing the position of pins and the proportions of rods and levers can be determined without difficulty. By a system of trials, a combination can be found best suited to obtain the desired results. Moreover, a model makes it possible to measure directly the slip of the link block along the link. This slip should be kept as small as possible to prevent rapid wear. It can be controlled to some extent by properly locating the link pins, by avoiding too short a link, and by choosing a favorable position for the reverse shaft.

The Zeuner diagram for a Stephenson gear embodies all of the principles of the Zeuner diagram for a simple valve, with certain additional ones which, while comparatively simple, sometimes cause confusion. It is only necessary to remember that there are two eccentrics and that their combined action is the same as one virtual eccentric; also, that in passing from a long to a short cut-off with open rods, the lead increases, hence the path of the virtual eccentric center must be a curved one. A practical example will make the construction of such a diagram clear.

EXAMPLE. Given a maximum valve travel of $5\frac{1}{2}$ inches, steam lap 1 inch, lead at full gear $\frac{1}{16}$ inch, and $\frac{R}{L}$ equal to $\frac{1}{4}$. Find the valve travel for 60 and 80 per cent cut-off, respectively.

SOLUTION. Construct the valve travel circle $A\ B\ C\ D$, Fig. 56, having a diameter of $5\frac{1}{2}$ inches. (The scale of the drawing is exactly $\frac{1}{2}$ size.) Draw the lap circle $T\ U\ V\ W$ and lay off the full gear lead $S\ T$. Lay off the angles $P\ O\ B$ and $Q\ O\ D$ equal to the angle through which the eccentrics must be turned in order to displace the valve by an amount equal to the lap plus the lead at full gear; or with slight error draw a perpendicular to $A\ C$ through

the point S and where it cuts the maximum valve circle, as at P and Q, will be the centers of the eccentrics sought. Two points of the locus of the virtual eccentric center have thus been established. In order to draw the locus, the amount of lead at mid gear must be known. By the construction of Fig. 53, it was shown that in shifting the link from the full gear position G_1H_1 to the mid gear position G_2H_2, the lead was increased by the amount G_1J, which can be measured directly from the drawing. In this problem assume that the

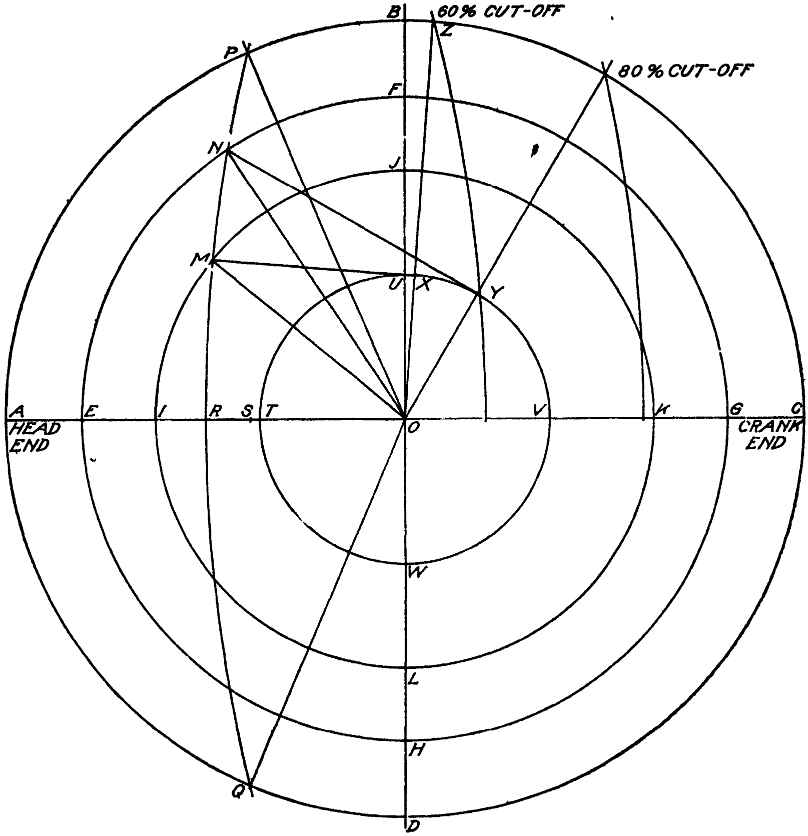


Fig. 56 Zeuner Diagram for Stephenson Gear

length of the eccentric blades is known and that by a construction similar to that in Fig 53, the lead at mid gear was found to be $\frac{3}{8}$ inch, or, in other words, in passing from full to mid gear the lead was increased $\frac{5}{16}$ inch. Knowing the lead at mid gear, lay off the distance TR equal to $\frac{3}{8}$ inch. The locus of the virtual eccentric center must pass through P, Q, and R and have its center on the line $A\,C$ extended. By trial, we find such a center and such a radius that the arc when drawn will pass through the points P, Q, and R. This arc is the locus of the virtual eccentric center when dealing with the head-end events. To find the events for the crank end, construct a similar arc on the right of the

vertical line $B\,D$. To obtain the valve travel for 60 per cent cut-off, first determine the crank position in the usual manner by locating the line $O\,Z$, remembering that $\frac{R}{L} = \frac{1}{4}$. Where this line cuts the lap circle, as at X, draw a tangent to the lap circle and extend it until it cuts the arc $P\,R\,Q$ at M. $O\,M$ is then the radius of the valve travel circle for 60 per cent cut-off. Construct the valve travel circle $I\,J\,K\,L$ with a diameter of 3½ inches, the required valve travel. In like manner, establish the point N, which determines the valve travel circle $E\,F\,G\,H$ for 80 per cent cut-off, the diameter of which is 4½ inches. By this same procedure, the valve travel for any cut-off

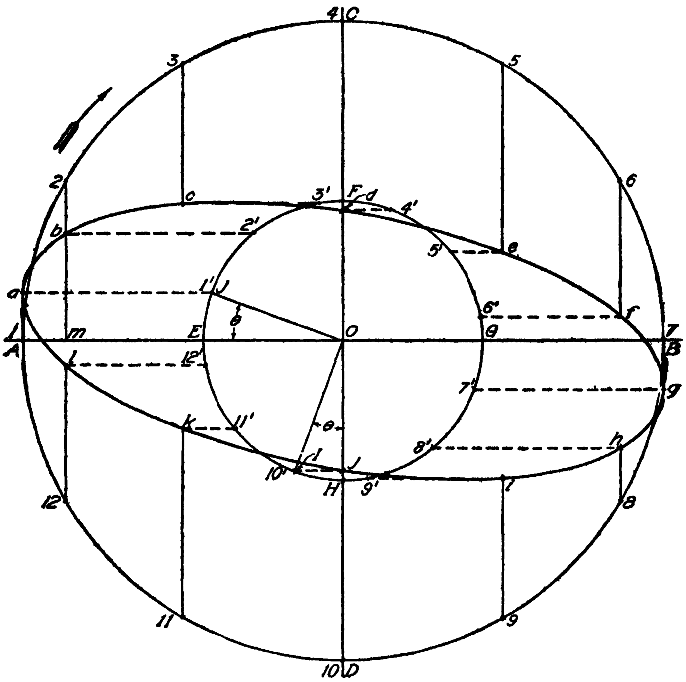


Fig 57 Valve Ellipse Diagram for Studying Valve Action

may be obtained. Having the valve travel circle established, all the events of the stroke may be found as has already been pointed out in the study of the Zeuner diagram.

Valve Ellipse Diagram. The valve ellipse diagram furnishes another method for studying the valve action, aside from that furnished by the Zeuner diagram. The valve ellipse has been used a number of years as a means for representing the relative positions of the valve and the piston.

The principle of its construction as applied to the arrangement of valve and rods, as shown in Fig. 55, is to draw lines at right angles to each other, one representing the travel of the piston, the other that of the valve. Thus, in Fig. 57, draw the circle *ABCD*, having a diameter equal to the stroke of the piston drawn to a predetermined scale. This circle represents the path of the crank pin center. Divide this circle into any number of equal divisions, in this case, twelve, at points *1*, *2*, *3*, etc. It is evident that if a line be drawn from any one of these points, as *2*, perpendicular to the line *AB*, that, neglecting the angularity of the connecting rod, the distance *A m* would represent the displacement of the piston as the crank moved forward from *A*. To allow for the angularity of the connecting rod, take a radius equal to the length of the connecting rod drawn

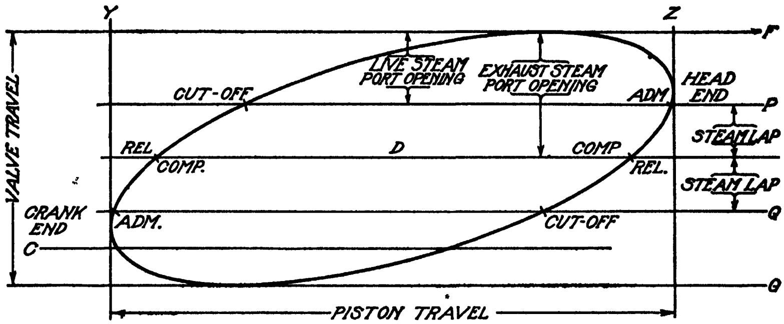


Fig. 58. Valve Ellipse Diagram Showing Information to be Obtained from its Analysis

to the same scale as that of the circle *A B C D* and sweep the arcs from the points *1*, *2*, *3*, etc., with the center on the line *A B* produced. Now representing the path of motion of the valve by the line *H F*, drawn perpendicular to *A B*, the eccentric position *O I*—which is located at the angle $(90-\theta)$ degrees behind the crank—is, for the sake of convenience in constructing the ellipse, located at *O J*. Having drawn the valve travel circle *E F G H*, begin at *J* and lay it off into the same number of equal parts as was done in the case of the crank circle. For the crank position *A*, the corresponding eccentric position is *J*, and hence, by projecting a vertical line and a horizontal line from *A* and *J*, respectively, the point *a* is located. In the same manner, the points *b*, *c*, *d*, etc., are located, thus completing the construction of the ellipse. The ellipse may have

different inclinations to the reference line, depending on conditions. This difference will be noted in comparing Figs. 57 and 58.

Thus far the discussion has dealt only with the construction of the ellipse. It is now proposed to point out what information may be obtained from the valve ellipse and for the sake of clearness, another figure is shown. After constructing the ellipse or having obtained it directly by an instrument specially constructed for the purpose, draw the reference line C in Fig. 58. Tangent to the ellipse, draw the lines F and G parallel to C. The distance between the lines F and G represents the travel of the valve. Midway between F and G draw the line D, the center line of the extreme travel of the valve. Assume that the valve is an ordinary plain slide valve having $1\frac{1}{8}$ inches steam lap and the zero exhaust lap, or line and line. Draw the lines P and Q $1\frac{1}{8}$ inches on each side of the center line D, and where these lines cut the ellipse determines the points where admission and cut-off occur for the two ends of the cylinder, as indicated in Fig. 58. Since there is no exhaust lap, the point where the line D cuts the ellipse gives compression and release for the two ends of the cylinder. In this case, the compression occurs on the head end at the same time that release occurs on the crank end, and *vice versa*. If the valve be given exhaust lap, it would be laid off in the same manner as the steam lap. Draw the lines Y and Z tangent to the ellipse and perpendicular to the reference line C. The distance between these two lines represents the length of the stroke of the engine drawn to scale. To find the per cent of the stroke at which any event occurs, it would be only necessary to drop a perpendicular to the center line D from the point on the ellipse corresponding to the event under consideration and obtain the percentage as previously explained. If the width of the steam port be known, it would be laid off from the lines P and Q toward the lines F and G as indicated. Assuming admission on the head end to occur as marked on the line P, it is evident from the portion of the curve contained between the lines F and P that at the beginning of the stroke the steam port was opened rather quickly and that cut-off occurred by the port being closed very slowly. During this time, the piston moved approximately three-quarters of the stroke. There being no exhaust lap or inside clearance, release occurred when the valve reached its central position. At the same

time that head-end release took place, compression began on the crank end, then followed crank-end admission, cut-off, release, and head-end compression, in regular order.

The valve ellipse has been largely used by steam railroad engineers and, as a result of the demand for such information as can be obtained from a consistent study of it, several devices have been invented for taking the ellipse directly from the engine. These devices consist of a drum the circumference of which is made proportional to the stroke of the engine. A sheet of paper is held on this drum by means of clips somewhat in the same manner as are found on the drums of steam engine indicators. This drum is mounted on a frame and when in use is placed in a convenient position above the crosshead or on the steam chest in such a position that its axis of rotation is perpendicular to the direction of the motion of the valve. The drum is rotated by means of a cord connection with the crosshead. Attached to the apparatus is a pencil which receives the same motion as that of the valve by means of a connection with the valve rod. Hence by the combination of the two movements, that is, of the drum moving with the piston and that of the pencil moving with the valve, the valve ellipse diagram is drawn.

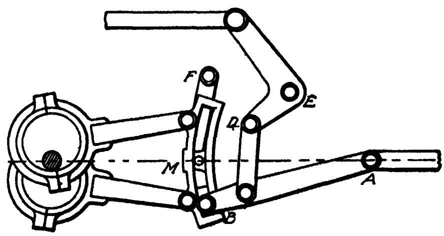


Fig. 59. Gooch Link Motion

From the study of the Stephenson gear, it is obvious that it is very flexible, and that it is readily adjusted to all irregularities of operation. Great care must be taken in its design in order that it may properly perform its work. Owing to the large number of parts and the size of same on large engines, it frequently gets out of alignment, its parts wear considerably and, on locomotives, the lubrication is sometimes difficult. On this account, it requires frequent attention in order that the best results may be obtained. All things considered, it is doubtful whether any other reversing gear gives as good a steam distribution as does the Stephenson gear when it is properly adjusted and operated.

Gooch Link. The Gooch link, illustrated in Fig. 59, has been extensively used on European locomotives, although it is gradually

being replaced by a type of valve gear known as the Walschaert, which will be described later.

The Gooch link has its concave side turned toward the valve instead of toward the eccentric. The radius of curvature of the link is equal to $A\ B$, the length of the radius rod. The link is stationary and the link block slides in the link. The engine is reversed by means of the bell-crank lever on the reverse shaft E which shifts the link block instead of the link, as is the case with the Stephenson. The link is suspended from its saddle pin M, which is connected by a rod to the fixed center F so that the link can move forward and back as the eccentricity is changed, or it can pivot about its saddle pin as the eccentrics revolve.

Since the radius of the link arc is equal to $A\ B$, it is apparent that the block can be moved from one end of the link to the other, that is, from full gear "forward" to full gear "back" without moving the point A, which is on the end of the valve rod. The lead then is constant for all positions of the block. The gear is more complicated than the Stephenson and requires nearly double the distance between shaft and valve stem.

RADIAL TYPE OF VALVE GEAR

In general, it would be desirable to have precisely similar steam distribution at each end of the cylinder, and it would often be of great advantage with a gear like the Stephenson if the cut-off could be shortened without changing any other event of the stroke. A Stephenson gear can be made to maintain equality of lead for both ends of the cylinder as the cut-off is shortened, but we have seen that in so doing, the lead of both ends is either increased or diminished according as the link is arranged with "open rods" or "crossed rods." Moreover, the compression is hastened by bringing the link to mid gear, all of which in many instances is undesirable.

This disadvantage of the Stephenson link motion led to the design of the so-called "radial valve gears," many of which are so complicated as to be impracticable, but all of which obtain a fairly uniform distribution of steam.

Hackworth Gear. The essential features of the Hackworth gear are indicated in outline in Fig. 60. In this figure S is the center of the shaft, and the eccentric E is set 180 degrees from the crank

SH. At the right-hand end of the eccentric rod EA is pivoted a block which slides in a straight, slotted guide. The guide remains stationary while the engine is running, but can be turned on its axis P to reverse the engine or to change the cut-off. P is a pivot which is located on a horizontal line through S in such a position that $D\,P$ is equal to EA. If these two distances are equal, A will coincide with P when the crank is at either dead point and the slotted guide may be turned from "full gear forward" (as shown in the figure)

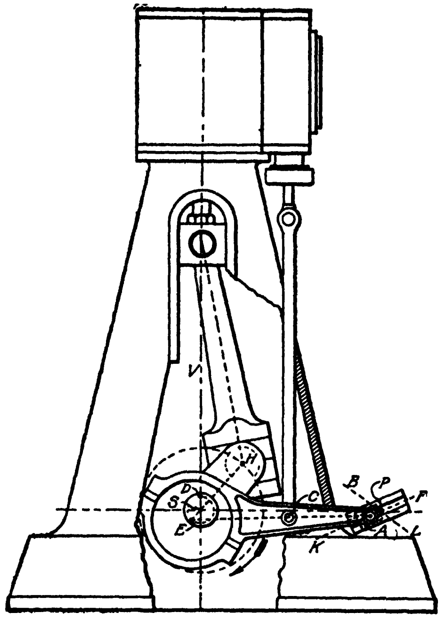


Fig 60. Diagram of Hackworth Radial Valve Gear

through the horizontal position to "full gear back" (as shown by the line BL) without moving the valve. It will be observed, therefore, that the leads are constant for all positions of the guide. The valve rod running upward from C connects with the valve stem, which it moves in a straight line. The valve stem is made just long enough to equalize both leads and, if the point C has been properly chosen, the two cut-offs will be very nearly equal for all grades of the gear.

A somewhat better valve action is obtained by slightly curving the slotted guide, with its convex side downward. This gear is sometimes used on marine engines and on small stationary engines.

Marshall Gear. The most objectionable feature of the Hackworth gear is, perhaps, the slotted guide, for the sliding of the block causes considerable friction and wear. The Marshall gear, shown in outline in Fig. 61, is designed to eliminate this feature. The point A moves in the desired path by swinging on the rod FA about F as a center. While the engine is running, the lever FP remains stationary, but can be turned on its axis P to reverse the engine or to change the point of cut-off. The pivot P is located precisely as

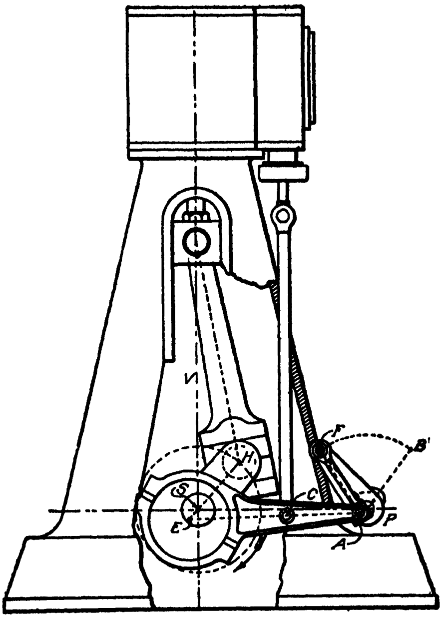


Fig. 61. Diagram of Marshall Radial Valve Gear

in the Hackworth gear, and the lever FP can be turned from "full gear forward" (as shown in the figure) to "full gear back" (as shown by the line BP), intermediate positions giving different cut-offs the same as with the Hackworth gear. Since FA is made equal to FP, the point A will always swing through P no matter where F may be and will coincide with P when the engine is on dead center. The leads for all positions of the gear, therefore, will remain constant, as in the preceding case.

The Marshall gear is sometimes made with the point C located on the right of A, on the line EA produced. In this case, if the same kind of valve is to be used, the eccentric E must move with

the crank instead of 180 degrees from it. The Marshall gear is frequently used on marine engines, the one eccentric being simpler than the two required by the Stephenson.

Joy Gear. The Joy radial gear, Fig. 62, is perhaps the most widely known, and is certainly one of the best radial gears. It is frequently used on marine engines and on some English locomotives. No eccentrics are used, the valve motion being taken from *C*, a point on the connecting rod. *H* is a fixed pivot supported on the cylinder casting. The lever *ED* has a block pivoted at *A*, which slides back and forth in a slotted guide, having a slight curvature, the concave side being toward the right. The guide and the lever *P F* are fastened to the reverse shaft *P* and, by means of a reverse rod leading off from *F*, can be turned from full gear forward, as shown, to full gear back, when the pin *F* moves over to the position *B*. Motion is transmitted to the valve stem by means of the radius rod *E G*.

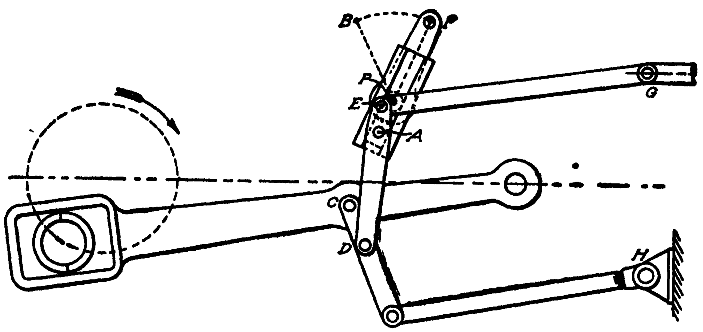


Fig 62 Diagram of Joy Radial Valve Gear

The proportions are such that when the crank is on either dead point, the pivot of block *A* coincides with *P*, so that the curved guide may then be set in any position without moving the valve; therefore the leads are constant. This gear gives a rapid motion to the valve when opening and closing and a more nearly constant compression than the Stephenson gear, and the cut-off can be made very nearly equal for all positions of the gear. Its many joints cause wear and its position near the crosshead makes a careful inspection of the crosshead and piston rather difficult while the engine is running.

Walschaert Gear. The Walschaert gear, Fig. 63, stands today as the best representative of the radial gear type. It has for many

years been very largely used on all the important European railroads. It has been used considerably in England and at the present time is being applied to a great many locomotives in America.

Analysis of Valve Motion. When the Walschaert gear is used, the valve receives its motion from two distinct sources, namely, from the crosshead and from the eccentric crank. In Fig. 64 the various parts of the gear are named. The crosshead connection gives to the valve a movement equal to the lap plus the lead, at the extremities of the stroke, when the eccentric crank is in its mid-position. The eccentric crank leads the main crank by an angle of 90 degrees for a valve having external admission, and follows the

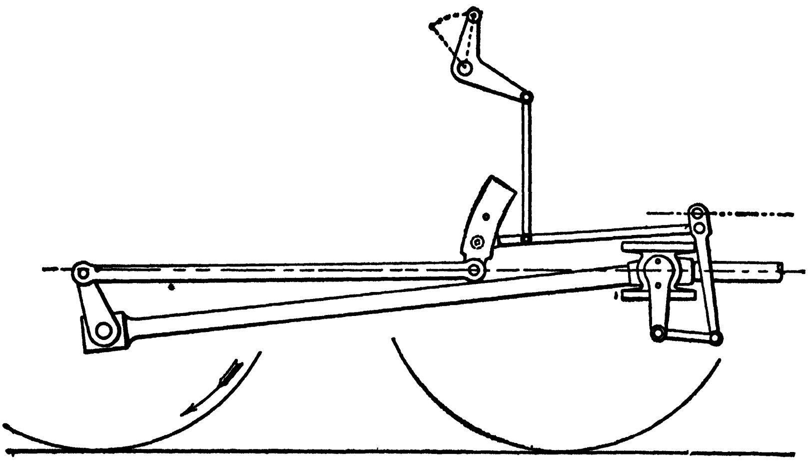


Fig. 63. Diagram of Walschaert Radial Valve Gear

main crank by 90 degrees for an internal admission valve. Locating the eccentric crank exactly 90 degrees in advance or behind the main crank is one of the necessary adjustments of the Walschaert gear. It is evident, therefore, that if an eccentric rod of proper length be attached to the eccentric crank and the valve through proper means, when the engine is on dead center, the valve would be in mid-position. However, in order to have economic operation, it becomes necessary to have some lead at dead center positions, hence the valve must be displaced by an amount equal to the lap plus the lead. Since the eccentric crank must be 90 degrees from the main crank, some other means must be used to

obtain the proper displacement and the method of accomplishing this on the Walschaert gear is one of its most distinguishing features. An attachment is made between the crosshead and the valve stem by means of a lever known as the combination lever, or. as shown in Fig. 64, the lap and lead lever.

In order to obtain the proper displacement of the valve when the engine is on dead center, the attachment of the combination lever to the crosshead and to the valve rod must bear a definite ratio to the stroke and valve travel. In other words

$$S : t \text{ as } L : V$$

or

$$V = \frac{Lt}{S}$$

in which S is stroke of piston in inches; t is twice the sum of the lap plus the lead in inches; L is distance between the crosshead connec-

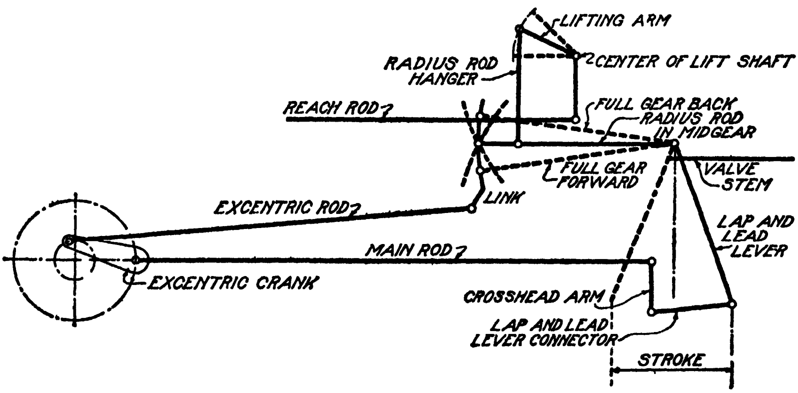


Fig 64. Diagrammatic Analysis of Walschaert Valve Gear Action

tion and that of the radius lever in inches; and V is distance between the connection of the radius lever and that of the valve stem in inches. The above expression holds good for either an inside or an outside admission valve.

When using an inside admission valve, the connection between the radius rod and the combination lever is made above the valve stem connection as shown in Fig. 64, that shown in Fig. 63 being the arrangement for an external admission valve.

Link Motion. We have thus far traced the movement of the valve, taking into consideration the crosshead and eccentric crank connection and omitting for the sake of clearness the link connection. It will be noted that the link is pivoted at the center. The link block is raised or lowered by means of the reverse lever and bell crank. The link block is connected to the radius rod, which has a length equal to that of the link; hence, when the engine is on either dead center, the link block can be raised from one extreme position to the other without moving the valve. Therefore this gear, if properly constructed, gives a constant lead for all positions of the reverse lever. The proper construction, suspension, and attachment of the link to its allied parts is a very important matter and one rather difficult to accomplish. The proper location of the attachment of the link to the eccentric rod gives the designer a great deal of trouble, in obtaining the desired action of the valve. In locating the longitudinal position of the link fulcrum, consideration must be given to the length of the eccentric rod, which should have a minimum length of three and one-half times the throw of the eccentric and should be made as long as the existing conditions will permit. It should be so located that the radius and eccentric rods are approximately of equal length. The point of connection between the link and the eccentric rod should be as near the center line of motion of the connecting rod as possible, making due allowance for the angularity of the rods. To accomplish this, it often happens that the throw would be excessive. In such cases, a compromise is necessary, the point of connection being raised above the center line of motion as the case demands. It has been found in designing this gear that these considerations require shifting the eccentric crank from one to two degrees, thus making the angle between the main crank and the eccentric crank 91 degrees or 92 degrees instead of 90 degrees, as theoretically required. The angle being increased by such a small amount does not affect the movement of the valve to any appreciable extent.

Adjustment of Gear. From the foregoing brief remarks, it is to be noted that in order to secure the best results, the design of the Walschaert gear requires very accurate work. No hard and fast rules can be laid down as how to secure the best design, for each case presents different problems. The best way to secure required results is to try out the design on a model.

The American Locomotive Company gives the following suggestions for adjusting the Walschaert valve gear:

(1) The motion must be adjusted with the crank on the dead center by lengthening or shortening the eccentric rod until the link takes such a position as to impart no motion to the valve when the link block is moved from its extreme forward to its extreme backward position. Before these changes in the eccentric are resorted to, the length of the valve stem should be examined as it may be of advantage to plane off or line under the foot of the link support which might correct the length of both rods, or at least only one of these should need be changed.

(2) The difference between the two positions of the valve on the forward and back centers is the lead and lap doubled and can not be changed except by changing the leverage of the combination lever.

(3) A given lead determines the lap or a given lap determines the lead, and it must be divided for both ends as desired by lengthening or shortening the valve spindle.

(4) Within certain limits, this adjustment may be made by shortening or lengthening the radius bar but it is desirable to keep the length of this bar equal to the radius of the link in order to meet the requirements of the first condition.

(5) The lead may be increased by reducing the lap, and the cut-off point will then be slightly advanced. Increasing the lap introduces the opposite effect on the cut-off. With good judgment, these qualities may be varied to offset other irregularities inherent in transforming rotary into lineal motion.

(6) Slight variations may be made in the cut-off points as covered by the preceding paragaph but an independent adjustment can not be made except by shifting the location of the suspension point which is preferably determined by a model.

Zeuner Diagram for Walschaert Gear. The Walschaert gear may be examined by the aid of a Zeuner diagram to the same limited extent as the Stephenson. The construction of the Zeuner for a Walschaert gear is somewhat easier than for the Stephenson because the locus of the virtual eccentric centers lie on a straight line, due to the constant lead. For example, take a maximum valve travel of $5\frac{1}{2}$ inches, a lap of 1 inch, and a lead of $\frac{3}{16}$ inch. In Fig. 65 (scale of drawing is exactly three-fourths size), the valve travel circle is $ABCD$, the lap circle $ILMN$. Lay off the given lead IF $\frac{3}{16}$ inch and through the point F erect a perpendicular line cutting the circle $ABCD$ at H and E, thus locating the two eccentric positions. Since there is a constant lead for any valve travel, the line HFE becomes the locus of the virtual eccentric centers. Assuming a cut-off of 80 per cent, locate the line OK and at J, the point where this line cuts the steam-lap circle, erect a perpendicular line

and extend it until it cuts the line $H F E$ at G. The point G is the extremity of the valve travel circle for 80 per cent cut-off, the radius

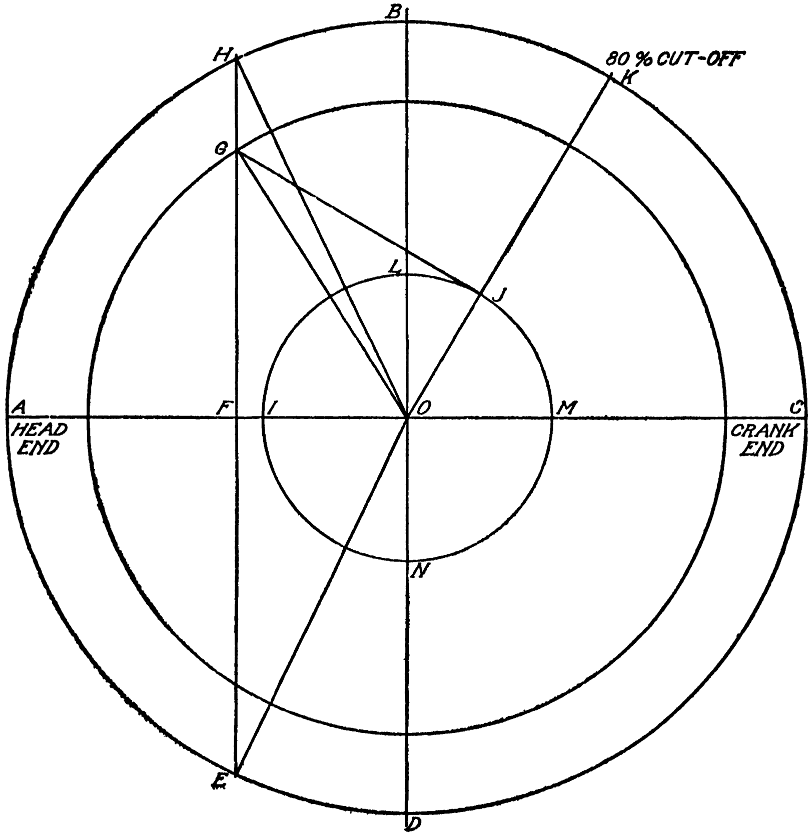


Fig 65 Zeuner Diagram for Walschaert Gear

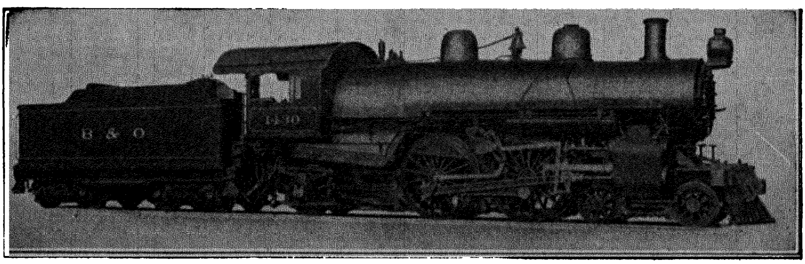


Fig 66. Application of Walschaert Valve Gear to a Passenger Locomotive

of which will be $O G$, $2\frac{3}{16}$ inches. In like manner, the valve travel can be obtained for any other point of cut-off.

Dimensions of Walschaert Gear Parts. For an engine such as is shown in Fig. 66, an approximate value of the various rods and levers may be taken as follows. By referring to Fig. 64 the location of the various parts can be determined more readily than in Fig. 66.

Main rod .	8′–0″	Radius rod . .	..3′–10″
Eccentric crank	1′–2″	Lap and lead lever (total) .	3′– 0″
Eccentricity	6½″	Lap and lead lever connector	1′– 2″
Eccentric rod	4′–6″	Crosshead arm . .	..1′– 0″
Link arc......	1′–10″	Stroke	. . ..2′– 0″

DOUBLE VALVE GEARS

It has been shown that a plain slide valve under the control of a gear that gives a variable cut-off, such as a shifting eccentric or a link motion, will not give a satisfactory distribution of steam at a short cut-off owing to excessive compression, variable lead, or early release. These difficulties are overcome in a measure by the use of the radial gear; and also by the use of two valves instead of one.

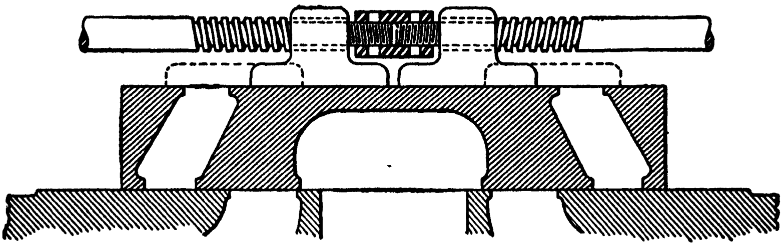


Fig 67 Section of Meyer Double Valve

The main valve controls admission, release, and compression; the other, called the cut-off valve, regulates the cut-off only, which may be changed without in any way affecting the other events of the stroke. This cut-off valve, sometimes known as the riding cut-off valve, may be placed in a separate valve chest, or it may be placed on the back of the main valve.

Meyer Valve. The most common form of double valve gear is the Meyer valve, Fig. 67. The cut-off valve is made in two parts and works on the back of the main valve. The two parts are connected to a valve spindle with a right-hand and a left-hand thread, so that their positions may be altered by rotating the valve spindle.

A swivel joint is usually fitted in the valve spindle between the

steam chest and the head of the valve rod, and the valve spindle is prolonged into a tail rod which projects through a stuffing box on the head of the steam chest, Fig. 68. The end of this tail rod is square in section and reciprocates through a small hand wheel, by means of which it can be rotated while the engine is running, whatever the position of the valve may be.

Each valve is under the control of a separate eccentric. The eccentric which moves the main valve is usually fixed, while the cut-off valve eccentric may be under the control of a governor. Since a slight compression is desired, the main valve is set to give late cut-off and this will also give late release and late compression, and allow a wide range of cut-off for the cut-off valve. With this gear, lead, release, and compression are entirely independent of the ratio of expansion, and the cut-off is much sharper, because the cut-off valve, when closing the ports, is always moving in a direction opposite to that of the main valve. The valve may be designed by means of the Zeuner diagram.

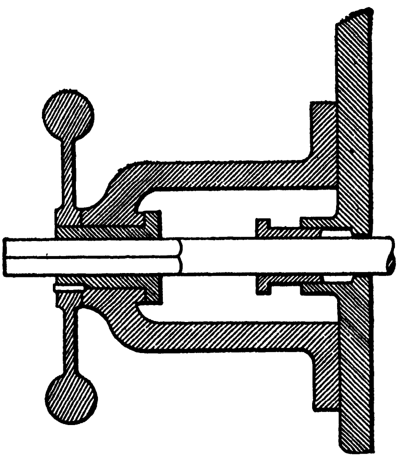


Fig. 68. Stuffing Box and Valve Spindle of Meyer Gear

Design by Zeuner Diagram. Let us design a Meyer valve having an eccentricity of 2 inches. Let the eccentricity of the cut-off valve be $2\frac{1}{4}$ inches and the relative travel of the cut-off valve in relation to the main valve be 3 inches. This will make the relative motion of the cut-off valve equivalent to the travel of a plain slide valve with an eccentricity of $1\frac{1}{2}$ inches. Let the outside lap on the main valve be $\frac{3}{4}$ inch, the lead $\frac{1}{32}$ inch, the compression 5 per cent of the stroke, and let the ratio of the length of the crank to connecting rod be 1 to 6.

In Fig. 69, draw $X\ O\ Y$, the main valve travel, equal to 4 inches. Lay off $X\ D$ equal to 5 per cent of 4 or 0.2 inches, and with a radius of 12 inches, and the center on $Y\ X$ produced, draw the arc $D\ H\ K$. $H\ K\ O$ is the crank position at compression on the head end; $C\ K\ O$, the crank position at compression on the crank end, is found in a

similar manner. Lay off $O\ I$ equal to the lap plus the lead, and draw the valve circle for the main valve through I and O with a diameter equal to its eccentricity of 2 inches. To do this, take a radius equal to 1 inch, and draw arcs from I and O as centers that shall intersect at B. B is the center of the valve circle and $O\ B\ E$ is the eccentricity, 2 inches. With E as a center, and with a radius equal to half the relative travel of the cut-off valve (in this $1\frac{1}{2}$ inches) draw an arc.

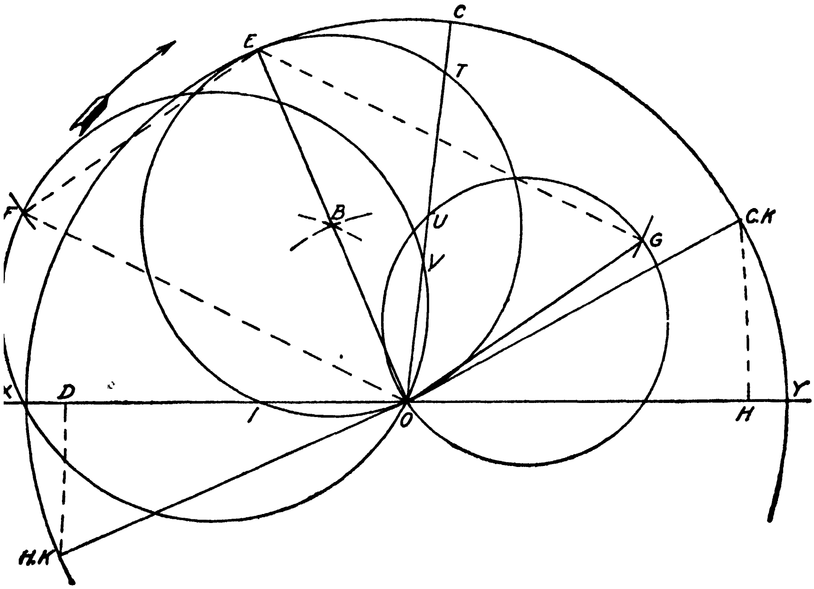


Fig 69 Zeuner Diagram for Meyer Valve Gear

With O as a center and with a radius equal to $2\frac{1}{4}$ inches, the eccentricity of the cut-off valve, draw another arc intersecting the first one at F. On $O\ F$ as a diameter, construct a valve circle. This valve circle will represent the absolute motion of the cut-off valve, independent of the motion of the main valve. This circle then will show the displacements of the cut-off valve from the center of the steam chest. With E as a center and with a radius equal to $F\ O$, draw an arc, and with O as a center and with a radius equal to $E\ F$, draw another arc intersecting the first at G. On $O\ G$ as a diameter, construct a valve circle. This circle will then represent the travel of the cut-off valve moving on the main valve. That is, it will represent the

displacements of the cut-off valve from the center of the main valve. This circle is not, properly speaking, a valve circle, and $O\ G$ is not an eccentricity, but simply represents the relative motion of the two valves. This can be proved by analytical geometry, but an inspection of the figure shows that this must be true.

Draw the crank line $O\ C$ at any position, cutting the valve circles at T and U and V. $O\ V$ represents the absolute displacement of the cut-off valve, that is, from the center of the steam chest, and $O\ T$ represents the displacement of the main valve. The relative displacement of the cut-off valve, that is, from the center of the main valve, will be the difference between $O\ V$ and $O\ T$, since both valves are moving in the same direction. By careful measurement it will be found that $O\ U = O\ T - O\ V$, and any arc as $O\ U$ on the auxiliary circle $O\ U\ G$ will correctly represent the displacement of the cut-off valve from the center of the main valve at the corresponding crank angle.

In Fig. 70 are shown $H\ K$ the crank angle at head-end compression, $C\ K$ the crank angle at crank-end compression, the main valve circle, and the auxiliary circle, all of which have been transferred from Fig. 69. In order to avoid confusion, the construction lines and all lines not essential to the figure are omitted.

Lay off on Fig. 70, $O\ I$ equal to the outside lap $\frac{3}{4}$ inch and draw the head-end lap circle $H\ E\ O$. It will intersect the valve circle for the main valve at L and M. Draw $H\ O$ through L, representing the crank position at admission (head end) and $O\ M\ H$ through M showing the crank position at cut-off. This gives the greatest possible cut-off. The cut-off valve may be set to give a much earlier cut-off than this, but of course, a later setting would be of no avail, for the port would be closed by the main valve at this angle. The crank line $O\ M\ H$ cuts the auxiliary circle at N_1 so that $O\ N_1$ ($1\frac{15}{32}$ inches) is the clearance of the cut-off valve. That is, the edge of the cut-off valve must be set $1\frac{15}{32}$ inches from the edge of the main valve port in order to cut off at this crank angle. The full lines of Fig. 66 show the cut-off valve placed in this position.

The intersection of $H\ K\ O$ with the lower valve circle gives the inside lap at the head end of the cylinder. This line comes so nearly tangent to the valve circle that the intersection can be determined only by dropping a perpendicular to $H\ K\ O$ from E_2. This cuts the

circle at P, and $O\ P$ equals the head-end inside lap $\frac{1}{32}$ inch, and $H\ E\ I$ represents the corresponding lap circle.

The crank position at compression on the crank end is $C\ K$, which cuts the upper valve circle at N_2, giving an inside lap for the

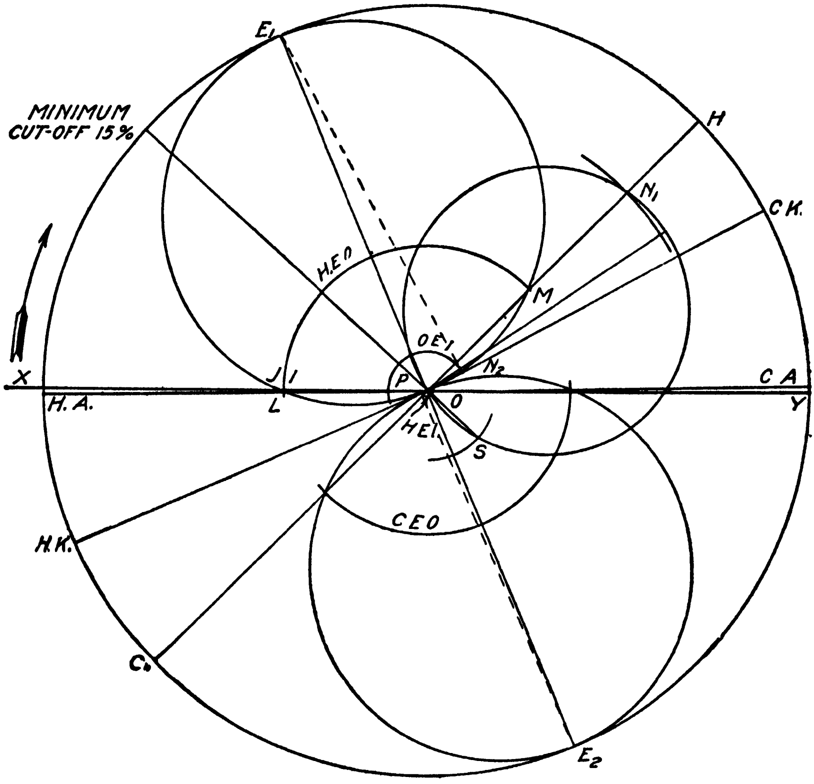


Fig 70 Zeuner Diagram Showing Additional Factors in Design of Meyer Valve

crank end, of $O\ N_2$, $\frac{13}{64}$ inch. To make this intersection more apparent, the perpendicular may be drawn from E_1, as previously explained.

Suppose that it was required that the minimum cut-off should be 15 per cent. Find the crank position at 15 per cent of the stroke in the same manner as the crank position was found at compression. Produce this line through O until it cuts the auxiliary circle at S. Then $O\ S$ equals the required lap $\frac{23}{64}$ inch for the cut-off valve in order to cut off at 15 per cent of the stroke. The dotted lines in Fig. 67 show the cut-off valve drawn in this position.

For a valve of this sort, the cylinder port should be 1¼ inches wide and the valve port 1 inch wide. Fig. 67 shows this valve laid out to scale, but as this process is in all respects similar to that described for laying out a plain slide valve, it will not be described in detail.

Shifting Eccentric Valve Gear. In addition to the valve gears already described, there is another class which receives a very large application, particularly in small and medium-power high-speed engines. This class of gear is what may be termed the shifting

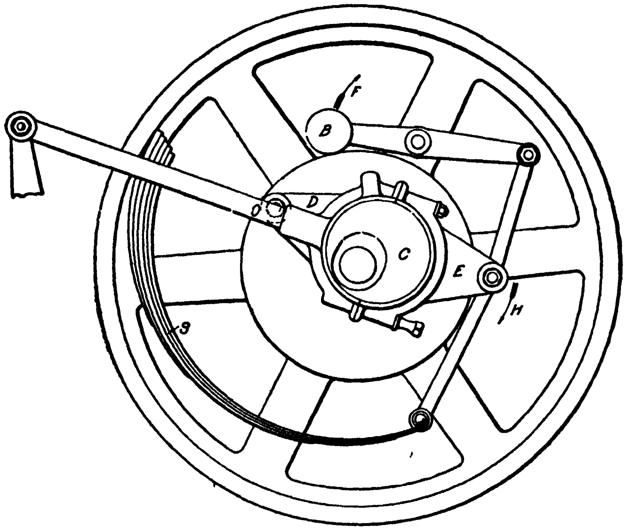


Fig 71 Diagram Showing Action of Straight-Line Type of Shaft Governor

eccentric gear. The valve itself is the ordinary flat slide or piston valve, and the valve stem, eccentric rod, and eccentric are the same as used in the common arrangement. The difference between the fixed and the shifting eccentric lies in the method of attaching the latter to the shaft and in the mechanism provided to move this eccentric from one position to another across the shaft. The general arrangement is illustrated in Fig. 71, which represents that used on the straight-line engine.

In this, *O* is the fixed pivot of the eccentric lever *E*, and *C* is the eccentric. The pin *H* of the eccentric lever is connected through a link to a leaf spring and through the other to a weighted lever *B*, as shown. When the engine is running, the position of the weight *B*

changes under different speeds and loads, and this change in position is transmitted to the eccentric. Since O is a fixed pivot, any motion of the eccentric lever E or eccentric C must be around O as a center. Consequently, when the eccentric position changes, its center will move in a path which is an arc of a circle with a center at O. The slot

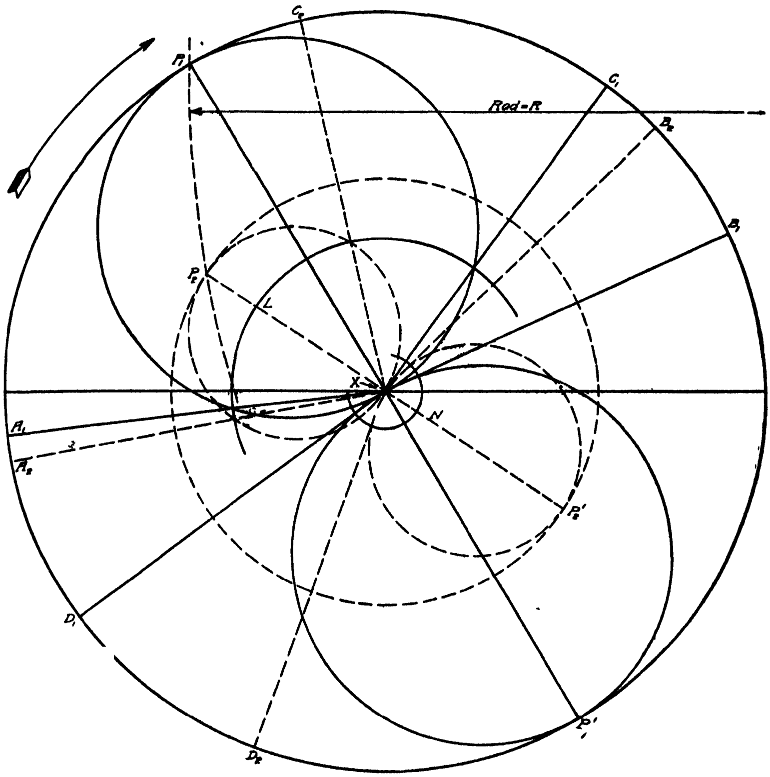


Fig. 72. Zeuner Diagram for Shifting Eccentric Valve Gear

in the eccentric is provided so as to enable it to move across the shaft to whatever position may be desired.

Two things are to be noticed for different eccentric positions. The first is that the eccentricity becomes less as the eccentric center moves along its path toward the shaft. The second is that the angular advance increases under this same condition. The first effect is to decrease the valve travel and the second to increase the angular

advance. The effect of the shifting of the eccentric on the motion of the valve is a combination of these two changes.

Fig. 30 shows the effect of changing the angular advance, and Fig. 31 shows the effect of changing the valve travel. The effect of both these changes acting together is shown in Fig. 72, which is a Zeuner diagram for a shifting eccentric valve gear.

Analysis of Zeuner Diagram. In Fig. 72 the full lines represent eccentric and crank positions at the point of maximum cut-off, and the dotted lines represent their positions corresponding to some earlier cut-off. A_1 is the crank position at admission; C_1 is its position at cut-off; B_1 is its position at release; and D_1 is its position at compression. P_1 and P_2 are the corresponding positions of the eccentric center, the subscripts referring to the maximum cut-off and to the earlier cut-off, respectively. The radius R is used to draw the path of the eccentric center and has the same length as the distance from the fixed eccentric lever pin to the center of the eccentric. The steam lap in the figure is XL and the exhaust lap is XN. The construction is made for the head end only, to avoid confusion due to the large number of additional lines required for that of the crank end. These laps remain the same for all positions of the eccentric.

Each of these two diagrams is made in exactly the same way as the ordinary Zeuner diagram and, if given the necessary data, the construction of the combined diagram should give no trouble. One point to bear in mind in drawing a Zeuner for this kind of valve gear is that the eccentric center for any position of the gear will lie somewhere along the arc described with the radius R.

An inspection of Fig. 72 shows that all events occur earlier with the earlier cut-off. However, they do not all continue for the same period, as was found to be the case when the angular advance alone was changed. This is because the valve travel is changed with the angular advance. The combined effect is that admission is advanced very slightly, cut-off is advanced a considerable amount, and release and compression are each advanced a moderate amount. Since the cut-off advances a greater amount than the release, the result is a greater expansion at earlier cut-off, which is more economical, within reasonable limits, in the use of steam. Also the increase in compression up to a certain point will make the engine run more smoothly by cushioning the piston better on the dead centers. Another effect of

earlier cut-off is a decrease of lead from that at maximum cut-off. This may or may not be an advantage, depending on the engine speed, construction of steam ports, amount of clearance, etc.

Thompson Automatic Valve Gear. The Thompson automatic valve gear, commonly known as the "Buckeye", belongs to the general class of double valve gears. Its principle of action involves certain ingenious points which make its study very interesting.

Two styles of valves of this type have been developed by the Buckeye Engine Company, namely, a flat valve and a round or piston

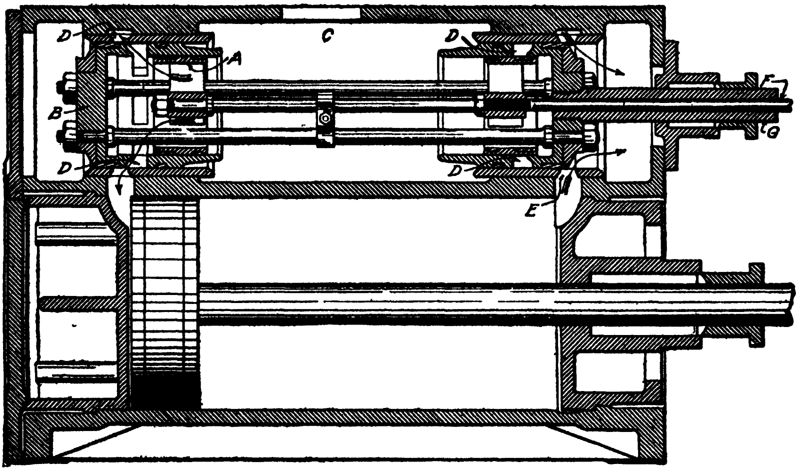


Fig. 73. Section of Cylinder and Valve-Gear Mechanism of Thompson Automatic Valve Gear

valve. While the essential features of the two valves are the same, the piston valve, illustrated in Fig. 73, is the simpler of the two and represents the latest practice. The cut-off valve A moves inside of the main valve B, and live steam entering at C passes through the cut-off ports D D in the main valve and is admitted to the cylinder, as these ports are alternately brought into coincidence with the cylinder ports E E, as shown by the arrows on the left. The exhaust steam is discharged at the ends of the main valve and does not come in contact with the valve except at the ends. It will be noted that the construction is such that the valve is at all times balanced.

The valve stem F of the cut-off valve passes through the hollow stem G of the main valve. Packing rings are used on both valves

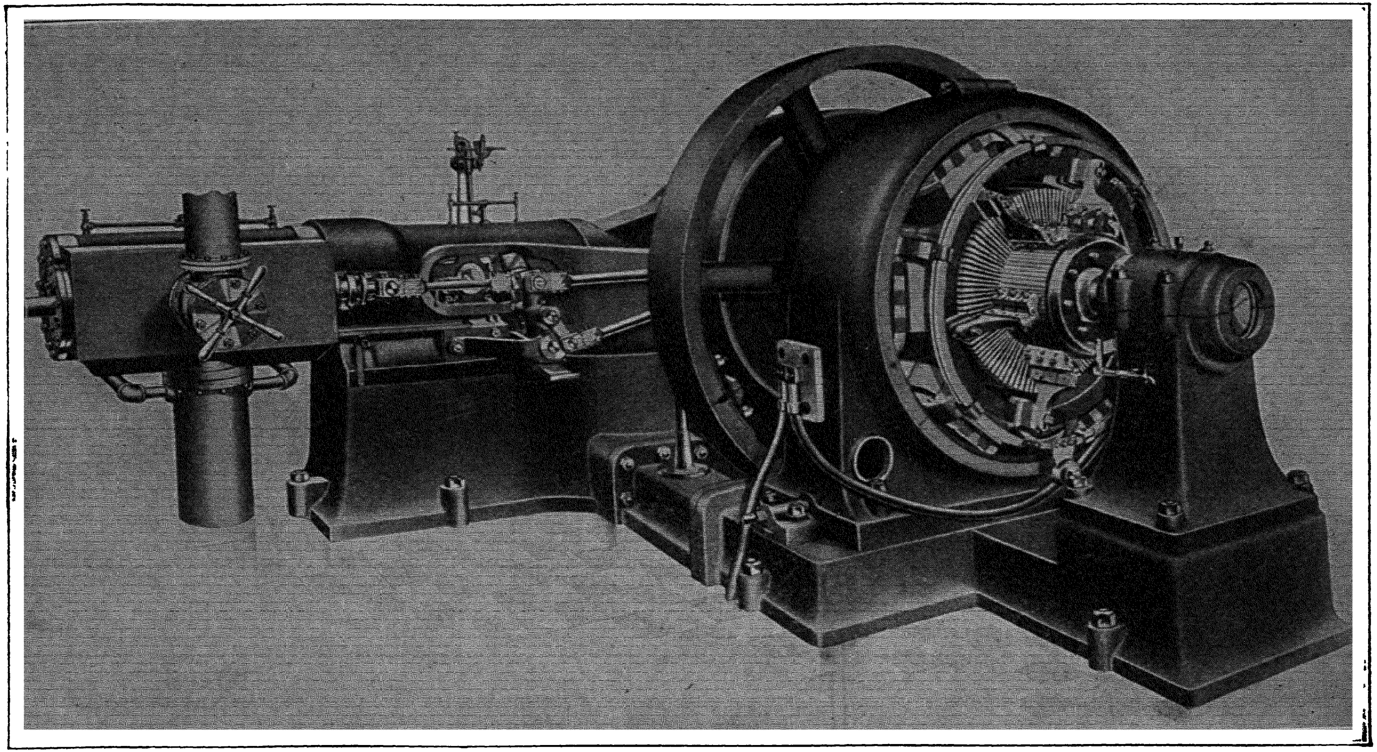


Fig 74 Thompson Automatic Valve Gear Applied to Buckeye Engine
Buckeye Engine Company, Salem, Ohio

to insure a steam-tight connection, and bridges are provided to afford a proper bearing for the rings in passing over the ports.

Fig. 74 shows the valve gear as applied to the engine. The operation of the gear can be better understood by referring to the line drawing, Fig. 75. The crank $A\ C$ is shown on the head-end dead center and running over. The eccentric D connects to the main valve stem through the eccentric rod $D\ M$, the joint M being guided by the rocker arm $H\ M$, pivoted to the engine frame at H. The cut-off valve is operated from the eccentric E by the eccentric rod $E\ F$, and the rocker arm $F\ K\ N$ is pivoted to the rocker arm $M\ H$ at

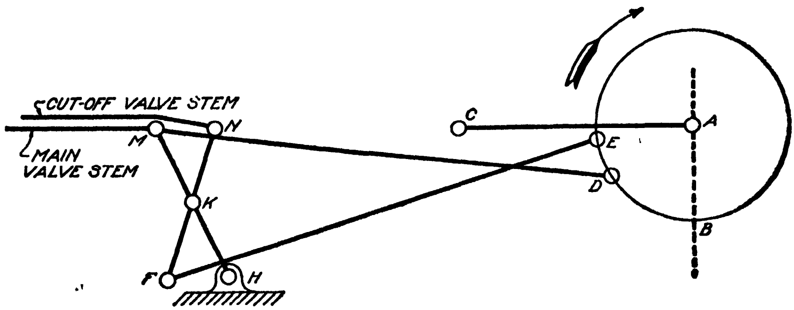


Fig 75. Diagram Showing Operation of Thompson Automatic Valve Gear

the point K. In the compound rocker arm, the arms $M\ K$, $K\ H$, $N\ K$, and $K F$ are all equal. On account of the valve under discussion being one having internal admission, it will be noted that the eccentric of the main valve follows the crank instead of preceding it, as is found in most cases.

Starting from the head-end dead center, suppose the crank, and likewise the eccentric, to turn through a small angle ϕ. This movement will cause the point M and the main valve to move to the left, a distance which we will call x, and the pivot point K will also move to the left a distance $\frac{x}{2}$. Now if F be considered as a fixed point, the movement of K, equal to $\frac{x}{2}$, causes the point N, and consequently the cut-off valve, to move to the left a distance x. The point F is not fixed, for while M is moving a distance x to the left, the rocker arm $F\ K\ N$, being pivoted to the rocker arm $H\ K\ M$ at the point K, will cause the point F to move to the left (due to the rotation of the eccen-

tric E) a distance which we will call y. This movement of F would cause N to move to the right a distance equal to y, provided the point K were stationary. Thus it will be seen that the point N and the cut-off valve are given a movement which is the resultant of two motions and is equivalent to $x-y$; and the relative movement of the two valves would be $x-(x-y)=y$. But y is the motion which would be given the cut-off valve by the eccentric E, independent of the other mechanism; *i.e.*, the construction is such that the cut-off valve moves on the main valve in much the same manner as an

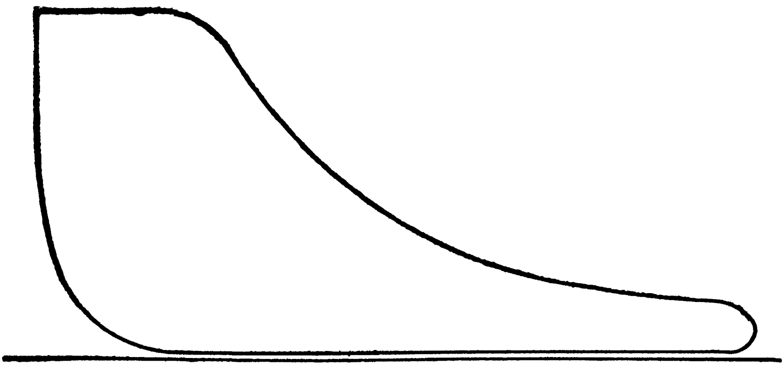


Fig 76. Indicator Diagram, Showing Effect of Application of Thompson Gear

ordinary plain slide or piston valve moves over stationary ports when connected to a constant throw eccentric through a reversing rocker arm.

The governor in moving the cut-off eccentric simply causes it to turn about the shaft, thus cutting off the steam earlier or later, according as the eccentric is advanced or moved back on the shaft. This action takes place without changing the cut-off valve travel or the relative movement of the two valves, since the throw of the two eccentrics is equal and constant. The Zeuner diagram for the Buckeye valve is worked out in a manner similar to that described for the Meyer valve.

Two important claims are made for this valve gear:

(1) On account of the valves moving in opposite directions at the instant cut-off occurs, cut-off is made very quickly, thus eliminating quite largely wiredrawing and giving an indicator diagram having a sharp turn at the point of cut-off, resem-

bling that given by a Corliss valve gear. This is illustrated by the diagram, Fig. 76.

(2) On account of the constant travel of the valves, they wear better than those that control the regulation by varying the valve travel.

This latter claim makes the gear particularly suited for the piston valve, since uneven wear or leakage is more liable to result from the packing rings if the valve movements are variable.

DROP CUT-OFF GEARS

The ordinary slide valve controls eight different events of the stroke, that is, admission, cut-off, release, and compression for both ends of the cylinder. A change in the setting of a plain slide valve that affects any one event on the crank end, let us say, will also change to a greater or less degree every other event of the stroke, on the head end as well as on the crank end; so that in setting a slide valve, the desired position for one event must usually be sacrificed in order to make the others less objectionable.

In order to provide a better distribution of steam than is possible with a single valve, some engines have four valves, two at each end of the cylinder. In horizontal engines, two valves are placed above the center line of the cylinder and two below, the upper being for admission and cut-off, the lower for release and compression. Since each valve controls but two events, a very satisfactory adjustment can be made and the extra complication and cost for large engines are more than overbalanced by the advantages gained, viz, a very much better distribution of steam; short steam passages and small clearances; separate ports for the admission of hot steam, and the exhaust of the same steam after expansion when its temperature has fallen; and finally the possibility of opening and closing the ports very rapidly, thus preventing wiredrawing. The small clearances, short ports, and separate admission and exhaust materially reduce the cylinder condensation, and thus effect a large saving in the steam consumption.

When four valves are used for high speeds, the motions of all must be positive, that is, they must be connected directly to some mechanism that either pushes or pulls them through their entire motion, but for speeds up to 100 revolutions or so, a disengaging

mechanism may be used, and the valves may shut off themselves, either by virtue of their weight or by means of springs or dashpots.

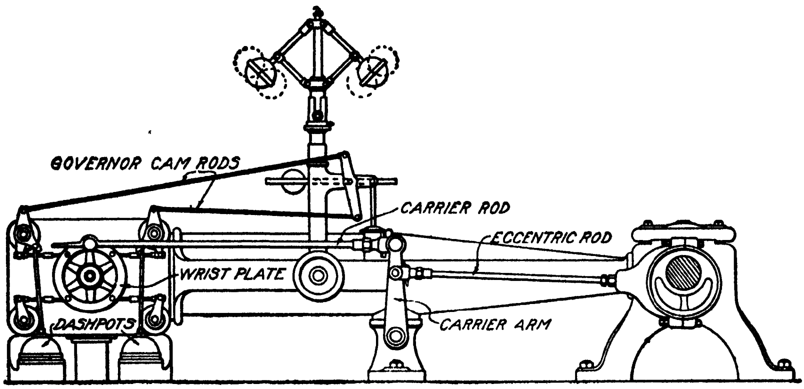


Fig 77. Diagram of Reynolds-Corliss Drop Cut-Off Gear

The valve is usually opened by means of links or rods moved by an eccentric and, at the proper point of cut-off, the rod is disengaged from the valve, which drops shut, hence the term "drop cut-off" gears.

Reynolds-Corliss Gear. The most widely known drop cut-off gear is the Reynolds-Corliss, Figs. 77 and 78. It is often referred to as the *Reynolds hook-releasing gear*. An eccentric on the main shaft gives an oscillating motion to a circular disk, called the *wrist*

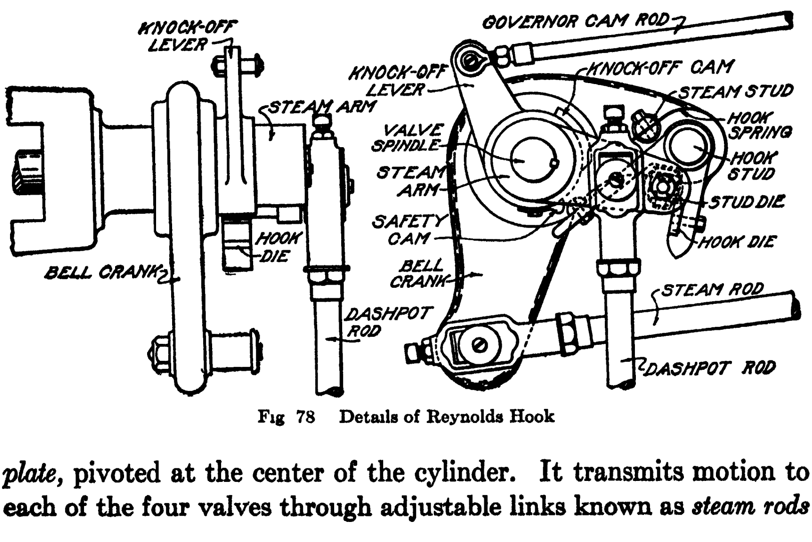


Fig 78 Details of Reynolds Hook

plate, pivoted at the center of the cylinder. It transmits motion to each of the four valves through adjustable links known as *steam rods*

or *exhaust rods,* according to whether they move the admission or exhaust valves.

The valves which are shown in section in Fig. 79 oscillate on cylindrical seats, and the position of the rods is so determined that they give a rapid motion to the valve when opening or closing, and hold it nearly stationary when either opened or closed.

The Reynolds hook is shown in detail in Fig. 78. The steam arm is keyed to the valve spindle which passes loosely through a bracket on which the bell-crank lever turns, and the spindle is packed to make a steam-tight joint where it enters the cylinder. Motion of the steam rod toward the right will turn the bell-crank lever and raise the hook stud. The hook (from which the gear derives its name), pivoted on this stud, has at one end a hardened steel die with sharp, square edges, and at the other end, a small steel block with a rounded face. As the hook rises, the hook die engages the stud which is fastened to the steam arm, and one end of the steam arm is thus raised. This turns the valve in its seat and admits steam. As the hook continues to rise, its stud moves in an arc above the valve spindle, and the round-faced block at its left-hand end strikes the knock-off cam, which causes the hook to turn about its stud and disengage the hook die from the stud die. In raising the steam arm, the dashpot rod is also raised and a partial vacuum is created in the dashpot. As soon, therefore, as the dies become disengaged, the dashpot quickly drops under the force of this vacuum, thus turning the steam arm and closing the valve. The striking of the left-hand end of the hook against the knock-off cam determines the point of cut-off by releasing the valve at that instant.

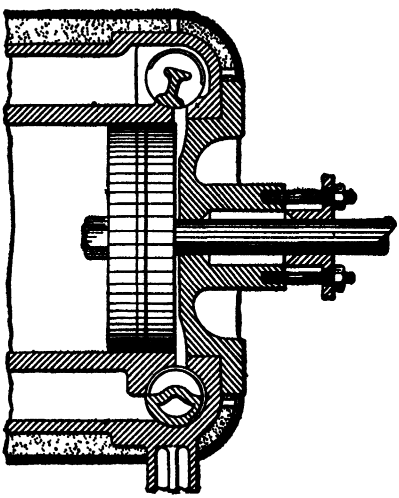


Fig 79 Diagram, Showing Reynolds-Corliss Valves in Section

This cam is a part of the knock-off lever to which the governor cam rod is fastened. Any action of the governor which would cause

the cam rod to move toward the right would cause this knock-off lever to turn on its axis, the steam arm, and consequently lower the position of the knock-off cam. This would cause an earlier contact between the cam and the end of the hook, and consequently an earlier cut-off. By lengthening or shortening the governor cam rod, the point of cut-off can be adjusted to suit the engine load without changing the speed.

There is a limit to this adjustment, for it can be shown that a Corliss gear operated by a single eccentric can not be arranged to cut off later than half-stroke. Suppose the eccentric is set just 90 degrees ahead of the crank. Then it will reach its extreme position just as the piston gets to half-stroke. If, by that time, the hook which was rising and opening the admission valve has not yet struck the knock-off cam, it can not strike it at all, for any further motion will cause the hook to descend to its original position, that is, its position at the beginning of the stroke; the hook will not disengage from the steam arm stud at all and the bell crank will return, closing the valve in the same manner in which it opened it. Cut-off will then take place near the end of the stroke, but it will not be the sharp cut-off produced by the sudden drop when the dies are disengaged.

If the eccentric were set less than 90 degrees ahead of the crank, the cut-off could be arranged to occur later than half-stroke, but this is decidedly impracticable, for with such a position of the eccentric, the action of the valves at release and compression is spoiled. When it is necessary to cut off later than half-stroke, as sometimes happens on low-pressure cylinders of compound engines, it may be arranged for by means of two eccentrics, one set *more* than 90 degrees ahead of the crank to operate the exhaust valves, and one *less* than 90 degrees ahead to operate the admission valves.

The safety cam, Fig. 77, is an important part of a Corliss gear. If for any reason the engine governor should fail to act, due, for instance, to the breaking of its driving belt, the governor would drop to its lowest position, supply more steam to the engine, and allow it to run away. The safety cam prevents this by moving so far to the right that it strikes the hook when it descends to pick up the steam arm. The hook is consequently turned toward the right and then lifted without engaging the stud die; the valve consequently remains closed and the engine stops.

Nordberg Gear. The Nordberg type of drop gear is designed for high speed and hard service. Instead of having its steam arm supported on the valve spindle, it is supported on a bearing formed by an extension of the steam bonnet, and the arm is provided with two hook dies instead of one. To eliminate side strains these two dies are connected on either side of the dashpot rod. The release is accomplished by means of an extension of the steam arm which rides in a slotted cam. The position of the latter is under control of the governor. The dashpot is also carried by the steam bonnet and is located above the gear. The spring type of dashpot is used to secure positive action at high speed.

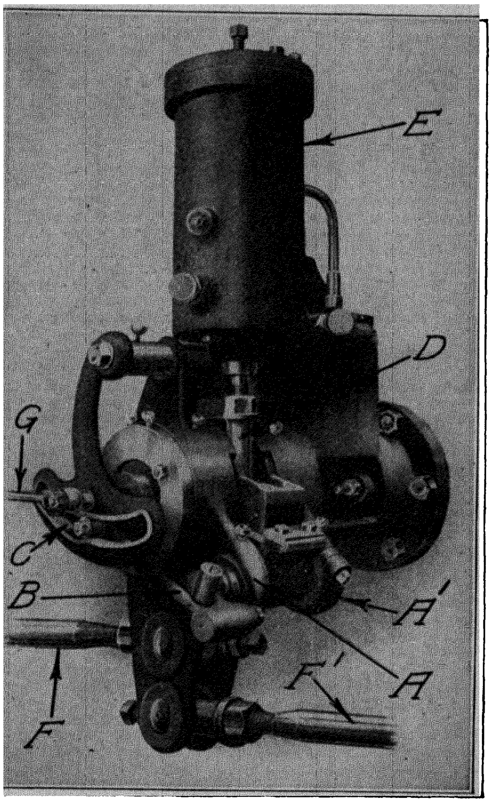


Fig 80 Nordberg Drop Gear
Courtesy of Nordberg Manufacturing Company, Milwaukee, Wisconsin

In Fig. 80, *A A* are the two parts of the steam arm to which the hook dies (not shown) are attached. *B* is the extension of the steam arm, and it carries at its left end a roller which works in the releasing cam *C*. When this roller strikes the off-set, shown in the cam, it raises the arm *B*, which disengages the hook dies and allows the dashpot rod *D* to snap the steam valve shut. *E* is the dashpot cylinder. *F F* are the driving rods, the one on the right being connected to the eccentric on the engine shaft, and the one on the left driving the valve gear for the other end of the cylinder. The rod *G* is connected indirectly to the governor and controls the point of cut-off by changing its position as the governor changes, according to the load on the engine.

Brown Releasing Gear. In addition to the Reynolds hook, several other devices are in use for opening and releasing Corliss admission valves. Among them, the Brown releasing gear, Fig. 81, may be noted. The steam rod and dashpot rod are arranged much the same as in the Reynolds gear. The governor cam rod operates a plate cam having a curved slot so shaped that it takes the place of both the knock-off and the safety cam of Fig. 78. The steam arm is keyed to the valve spindle and carries at its lower end a steel die which is free to slip up and down a small amount. The part of this gear corresponding to the Reynolds bell crank becomes a straight rocker pivoted at its middle; and the part corresponding to the Reynolds hook has at one end a die which engages the die of the steam arm, and at its other end a roller running in the curved cam slot. This hook is really a bell-crank lever with arms that are not in the same place. The bearing on which it turns is carried on the lower end of the rocker and, therefore, is equivalent to a movable pivot similar to the hook stud of the Reynolds gear.

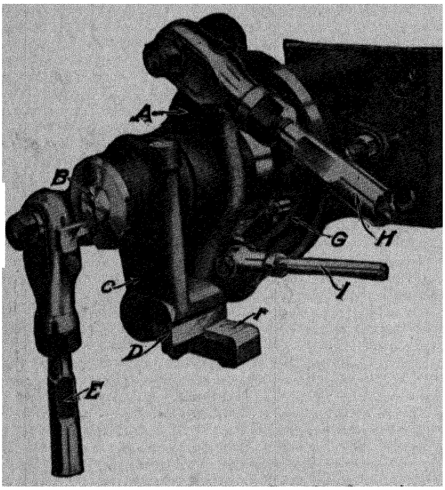


Fig 81. Brown Releasing Gear for Operating Corliss Admission Valves

In the position shown, the dies are engaged. Motion of the steam rod toward the right will move the lower end of the rocker toward the left, and consequently turn the valve spindle in a right-hand direction. This will open the valve and at the same time raise the dashpot rod. Meanwhile, the roller is moving toward the left in a circular part of the cam slot, the center of which is at the center of the valve spindle. This causes the steam arm and the bell-crank lever, which has the roller at one end, to move in such a way that there is no relative motion between them. As soon, however, as the roller comes to the point where it is forced to move out of this circular path and move farther from the valve spindle, both arms of the bell-crank lever are turned downward, the dies become disen-

gaged, and the dashpot closes the valve. The slight up-and-down motion of the steam-arm die allows it to rise, while the hook die passes underneath when returning to re-engage for the next stroke. The makers claim that this gear permits a much higher speed than is possible with other Corliss gears.

Greene Gear. Another well-known drop cut-off gear is the Greene, Fig. 82. The valves are of the gridiron type, sliding on horizontal seats, the admission valves parallel to the axis of the cylinder, and the exhaust valves at right angles to the axis of the cylinder and just below it. *A A* are rock shafts turning in fixed bearings. *B B* are the admission valve stems. *C* is a slide bar, receiving a reciprocating motion from an eccentric. *T T* are tappets connected to the slide bar. They move to and fro with the slide bar and can also move independently up and down. They are made fast at their lower end to the gauge plate *D*, which slides through the guide *E*. The guide *E* is made fast to the governor rod *F* and through this means can be raised or lowered, thus regulating the height of the tappets.

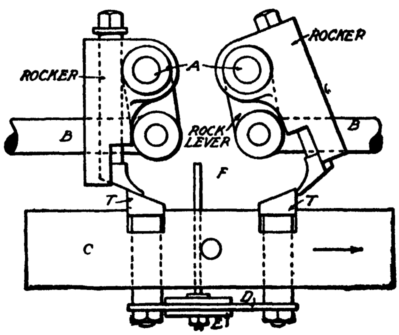


Fig. 82 Greene Drop Cut-Off Gear

As the slide bar moves toward the right, the right-hand tappet is brought into contact with the toe of the rocker, causing it to turn on its bearings and move the rock lever and the valve stem *B* toward the right, thus opening the admission valve. Since the tappet moves in a horizontal direction, while the toe of the rocker moves in an arc, it will be seen at once that they will soon become disengaged and release the valve, which is at once closed by a dashpot (not shown in the figure). If the governor raises the tappets, cut-off will be later. A nut at the bottom of the governor rod allows a proper adjustment of the guide and guage plate. As the slide bar *C* moves toward the right, the left-hand tappet comes in contact with the heel of the left-hand rocker and, both being beveled, the toe of the rocker rises in its socket, allowing the tappet to pass under. It then falls by its own weight and is ready to engage the tappet on its return and open the valve. In this gear, the disengagement of the valve throws

no load whatever on the governor, a distinct advantage over the Corliss gear. The action of the exhaust valves is not shown in the cut.

Sulzer Gear. The Sulzer gear is a drop cut-off widely used in Europe. The valves are of the poppet type, lifting straight from conical seats, so that there is no friction. They are usually placed vertically above and below the cylinder axis and are operated by eccentrics from a shaft geared to the main shaft. The admission valves are lifted from their seats by suitable levers, then released by a device equivalent in action to the Reynolds hook and are quickly closed by the action of springs.

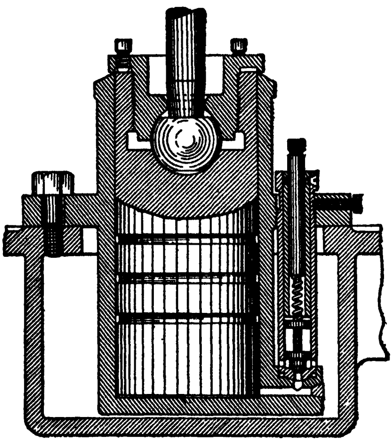


Fig 83 Form of Vacuum Dashpot for Closing Admission Valves

The exhaust valves of all drop cut-off gears are comparatively simple in their operation, and both in opening and closing are moved by the direct action of the exhaust rods.

A common form of vacuum dashpot for closing admission valves is shown in Fig. 83. The rod coming down from the steam arm makes a ball-and-socket joint with the dashpot piston. The dashpot is often let down into the engine frame, as shown. When lifted, the piston produces a partial vacuum underneath it so that it tends to drop quickly as soon as the valve gear is released. On some of the largest modern engines where the valves are very heavy, steam-loaded dashpots are used; that is, the dashpot piston has steam pressure on one side, and an air cushion on the other prevents it from striking the bottom of the dashpot.

CORLISS VALVE SETTING

The setting of a Corliss valve gear is a much longer process than the setting of a plain slide valve, but is nevertheless a comparatively simple matter, for the various adjustments are nearly all independent of one another. In gears like that shown in Fig. 77, the length of both the eccentric rod and the carrier rod are usually adjustable, and the former should be of such length that the carrier

arm swings equal distances on each side of a vertical line through its pivot, and the carrier rod should be adjusted until the wrist plate oscillates symmetrically about a vertical line through its pivot. Nearly all Corliss engines have one mark on the wrist plate hub and three on the wrist plate stand, as shown in Fig. 84, and the wrist plate should swing so that *A*, the mark it carries, moves from *C* to *D*, but not beyond either one. When *A* is in line with *B*, the wrist plate is in mid-position. The valves are then not in their exact mid-position, but it is customary to regard them as being in mid-position, and to speak of the laps as the amount which the port is covered by the valve when the wrist plate is in mid-position.

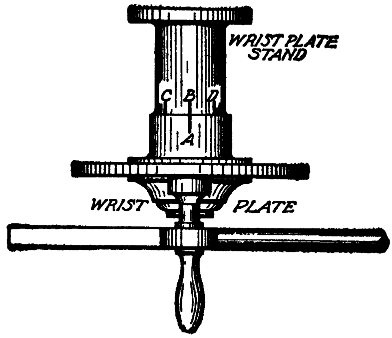


Fig 84. Corliss Wrist Plate for Adjusting Carrier Rod

Adjusting Steam Lap. To set the valves, remove the bonnets or covers of the valve chambers on the side opposite the gear. The ends of the valves are circular, but on their inside the cross section is as shown in Fig. 85. On the end, in line with the finished edge of the valve and on the seat in line with the edge of the steam port, are marks, as shown in Fig. 85. When these marks coincide, the valve is either just opening or just closing, and when in any other position, the amount of opening or the amount by which the port is closed is shown directly by the distance between the marks. Block the wrist plate in mid-position, hook up the admission valves, and adjust the length of the steam rods by means of the right and left threads provided for the purpose, until the ports are covered by the amount of lap indicated in Table II, opposite the given size of engine.

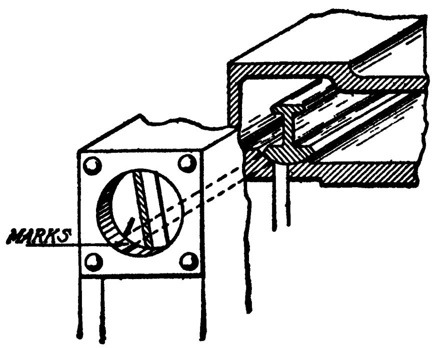


Fig 85 Diagram Showing Method of Adjusting Steam Lap for Corliss Valves

TABLE II

Standard Lap and Clearance Values

Diameter of Cylinder Inches	Steam Lap Inches	Exhaust Clearance Inches
12	$\frac{1}{4}$	$\frac{1}{32}$
14 to 16	$\frac{5}{16}$	$\frac{1}{32}$
16 to 22	$\frac{3}{8}$	$\frac{1}{16}$
22 to 28	$\frac{7}{16}$	$\frac{3}{32}$
28 to 36	$\frac{1}{2}$	$\frac{1}{8}$
36 to 42	$\frac{5}{8}$	$\frac{5}{32}$

Adjusting Exhaust Clearance and Lead. Next adjust the exhaust rods until the exhaust ports are open an amount equal to the clearance given in Table II. Set the engine on its head-end dead point, hook the carrier rod onto the wrist plate and in the direction in which the engine is to run, turn the eccentric enough to open the head-end admission valve by a proper amount of lead; then the eccentric will be $(90+\theta)$ degrees ahead of the crank. The proper amount of lead will depend upon both the design of the gear and the speed at which the engine is to run; and may vary from $\frac{1}{32}$ inch for small engines to as much as $\frac{5}{32}$ inch or $\frac{3}{16}$ inch for large engines and those of higher speed. When the proper amount of lead has been obtained, fasten the eccentric on the shaft by means of the set screw and make sure by trial that the wrist plate moves to its extremes of travel. The dashpot rods must be adjusted so that when the dashpot piston is at its lowest position, the hooks, Fig. 78, descend just far enough for the hook dies to snap over the stud dies with about $\frac{1}{32}$ inch to $\frac{1}{16}$ inch to spare, depending on the size of the gear.

Adjusting Cut-Off. To adjust and equalize the cut-off, lift the governor to about the position that it will occupy when running at normal speed, and put a block under the collar to hold it in this position. First, set the double lever at the right of the governor cam rods, so that it makes approximately equal angles with each rod, and then turn the engine over by hand until the piston has moved to the desired point of cut-off. Adjust the proper cam rod until the knock-off cam strikes the hook and allows the valve to close, then turn the engine to the point of cut-off on the other stroke and adjust the other cam rod in a similar manner. Now set the governor

in the lowest position to which it could fall if there were no load on the engine, and set the safety cams so that in this position the hook can not open the valve. A latch is provided, on which the governor can be supported slightly above its lowest position, so that the valves can be opened by the hook when starting the engine. As soon as the engine speeds up, this latch must be moved aside, so that if the governor fails to act, it can drop to its lowest point, otherwise this latch would hold it just high enough so that the safety cams could not act.

When Corliss gears are set, as here described, the position of the eccentric may not be quite right, due to an incorrect estimate of the amount of lead required. The error is likely to produce faulty release and compression as well as poor admission, but it can not be very serious, and the engine will turn over with its own steam, so that indicator diagrams may be taken. The final adjustments can then be determined from an examination of the diagrams.

VALVE GEAR TROUBLES AND REMEDIES

Importance of Keeping Valve Gear in Condition. The valve motion, or valve gear, is primarily responsible for the correct steam distribution in all steam engines. It follows, then, that in order to maintain efficient operating conditions the valve gear should receive constant careful attention in order that any irregularities which may develop can be detected at once and the fault corrected. A great many of the different gears used in American practice are described earlier in this text. In a number of cases the methods used in adjusting the gear and setting the valve have been given. For this reason the matter presented under this heading will be principally a discussion of the methods to be followed when trouble develops under conditions of service.

Familiar Types. Of the many different types of valves gears described in the preceding pages, perhaps the most familiar ones are included under the following heads:

(1) The direct-acting duplex pump valve gear
(2) The plain D-valve or piston valve gears of the simple steam engine
(3) The Corliss engine valve gear
(4) The Stephenson link motion valve gear
(5) The Walschaert radial valve gear

DUPLEX PUMP VALVE GEAR

Description. A great variety of valve gears are used in direct-acting steam pumps. The most common form, and in many respects the most reliable, is that illustrated in Fig. 50, in the text "Steam Engines". A pump such as shown in the illustration is nothing more than two pumps combined. In this particular design the motion of the piston rod of each pump is made use of in operating the valve of the other. In such a gear the only part which is made adjustable is the length of the valve rod. It is easily seen therefore that the setting of the valve is a comparatively simple matter.

Possible Troubles. After such a pump has been in service for some time, it may be necessary to dismantle it preparatory for removal to a machine shop for repairs. An accident may happen in which one of the operating arms, which are usually made of cast iron, becomes broken. In either case the operating engineer should be in a position to readjust the parts and properly set the valves after the necessary repairs have been made.

Setting Valves. The valves of such a pump are usually of the D-type, but piston valves could be used to advantage if desired. In setting the valves the general procedure should be as follows:

Preparation. Remove the steam chest or valve chest covers so that the movement of the valve relative to the ports can be measured.

Measuring Valve Travel. Move each of the pistons as far as it will go against the head in one direction and make a pencil mark on the seat of each valve at its edge, the farthest from the center of its travel. Now move the pistons against the other cylinder heads and make pencil marks on the valve seats, but on the other edge of the valves. The marks on the valve seat indicate the travel of its valve, which should be symmetrical with the ports.

Equalizing Valve Travel. If the travel is unequal, relative to the steam ports, the valve stem should be adjusted until the valve overtravels each steam port by the same amount. When the travel of each valve has been equalized, the setting may be considered finished and the valve chest covers may be replaced and parts connected.

Variation in Conditions. In the adjustments just explained it is assumed that the gear was originally proportioned properly so as to cause the valve of each pump to open early enough to

prevent the pistons from striking the cylinder heads. It sometimes happens that the closing of the valves on the water end of the pump is such as to require a slightly different setting from that explained above in order that the pistons may be reversed to prevent striking. When such is the case the necessary adjustment should be made. Each individual case will probably need different treatment and cannot be anticipated.

PLAIN SLIDE VALVE GEAR OF SIMPLE STEAM ENGINE

Types. As previously explained the plain slide valve gear of the simple steam engine is the simplest of all steam engine valve gears. On account of its simplicity it is less liable to get out of adjustment or meet with an accident which would totally disable its action. The essential elements of this gear are shown in Figs. 3 and 4, 6 to 10, 20 and 21. It is usually found constructed in one of three forms: (1) the form in which the valve receives its motion directly from the eccentric; (2) the form in which the valve receives its motion through the medium of a rocker arm in such a manner that the valve rod and eccentric rod move in the same direction; and (3) the form in which the valve receives its motion through the medium of a rocker arm in such a manner that the valve rod and eccentric rod move in opposite directions.

Use of Rocker Arms. The use of the rockers mentioned in the forms (2) and (3) is usually made necessary by the design of the engine, which is such that the valve stem and eccentric cannot be placed so they will be in the same straight line. In the adjustment of such gears it is essential that the rocker be so located that it will vibrate equally on either side of a vertical line drawn through the fulcrum point. If the rocker is not adjusted as directed, the valve will receive a motion which may be faster in one direction than the other even though its travel is equalized. When such conditions exist it is impossible to secure a valve setting which will give a correct distribution.

Slipped Eccentric. A trouble which is sometimes experienced when the engine is in operation is caused by the eccentric becoming loose on the shaft and slipping around in such a way as to reduce the power of the engine or perhaps cause the engine to stall. This usually happens in engines where the eccentric is held

in position by means of a set screw. If a key is used the trouble is seldom experienced.

Method of Correction. When it does happen that the eccentric slips, the operating engineer can get a setting which, while not absolutely correct, will permit the engine to be operated with but little loss of time by following the directions here given:

First, set the engine on the head or crank end dead center, by inspection, with as much accuracy as is possible under the circumstances. *Second,* turn the eccentric around the shaft in the direction the engine is to run until steam will just begin to blow out at the cylinder drain cock on the end in question when the throttle is opened slightly. When this position is found, tighten the set screw in the eccentric temporarily. *Third,* turn the engine over to the other dead center and see if steam blows from the corresponding cylinder drain cock with the same degree of freedom. If it does the eccentric may be said to be in the correct position and the set screw may be securely tightened. When this is done the engine will be ready to again assume its duties. At the first opportunity the valve setting should be carefully checked by one of the methods described earlier in this text.

Increasing Power Capacity. It frequently happens in small plants using a plain slide valve engine that additional machines will be added from time to time until the engine finally becomes overloaded under ordinary conditions of operation. Under such circumstances the operator is asked to devise means of increasing the power delivered by the engine. This can be accomplished in one of the following ways: (1) by increasing the speed of the engine; (2) by increasing the pressure carried by the boiler; (3) by increasing the point of cut-off; and (4) by the combination of any two or all of the above methods.

Importance of Boiler Capacity. An examination into the methods given above reveals the fact that in every case additional load will be placed on the boiler. If the boiler capacity is sufficient to carry the additional load, then the problem can be solved, otherwise it cannot.

Increasing Speed. If the power is increased by increasing the speed of the engine to any very great degree, it will be necessary to change the size of the belt pulleys on the engine and line shaft in order not to disturb the speed of the machines.

Increasing Boiler Pressure. In increasing the power by increasing the boiler pressure, no changes are necessary unless it is thought advisable to replace any or all of the high-pressure steam pipe and fittings with extra heavy grade.

Lengthening Point of Cut-Off. If it is desired to increase the power by lengthening the point of cut-off, this can be accomplished by removing the valve and planing off the ends, thus reducing the steam lap the desired amount to give the increased cut-off. It is very essential to remove the same amount from each end of the valve, otherwise the steam lap would be different for each end. If the engine was originally cutting off at one-half stroke and it is desired to have the cut-off increased to three-fourths stroke, the amount of metal which should be removed to give the desired condition can easily be determined by drawing a Zeuner diagram from the valve in question. When the valve is finally reconstructed and placed in position in the steam or valve chest, it will be necessary to change the angle of advance of the eccentric in order to secure the proper amount of lead. To secure the proper setting it would be advisable to follow the directions given earlier in this text.

Use of Double Valve. As has been previously pointed out, the plain D-valve possesses certain objectionable features in the matter of steam distribution which is partially overcome by the use of a double valve. The Meyer valve is perhaps the most common form of double valve, a description of which is given on pages 73 to 78 of this text.

Setting Meyer Valve. In setting the Meyer valve, the main valve is set in the same manner as the ordinary simple D-valve. This main valve controls the admission, release, and compression points, while the riding, or secondary, valve controls the point of cut-off. Having correctly set the main valve, connect the riding valve to its eccentric and adjust the rods so that its travel is equal on each side of its central position, in exactly the same way as directed for the simple D-valve. When this is done, place the piston at the point where cut-off is desired and rotate the riding eccentric in the direction the engine is to run until a point is reached where the valve is just cutting off. When this point is reached fasten the riding eccentric to the shaft. Next place the

piston at the same relative position on the other stroke, and, if cut-off is just occurring, the valve may be said to be correctly set and the riding eccentric securely fastened. If cut-off does not occur at the same point on each end, make adjustments of the eccentric and valve rod until the cut-off points are equalized.

Pounding or Knocking. The question of pounding or knocking is discussed in "Steam Engines", but since this is frequently caused by improper valve setting, it seems well to give this troublesome matter a brief consideration.

Indications of Faulty Valve Action. If an annoying pound is heard which is difficult to locate, it is probably due to an improperly set valve. If this is the real cause of the trouble, it will be easily shown by indicator cards taken from the engine when under regular operating conditions. If the pound is due to valve action it will be revealed in the indicator cards in one or all of the following three things: (1) by compression beginning so early that the compression pressure exceeds the steam line pressure, thus causing the valve to be raised from its seat until the admission point is reached when the valve is forced to its seat with a "slam"; (2) by admission occurring so late that the lost motion first "runs out" and is then taken up after steam has been admitted; and (3) by the unequal distribution of power between the two ends of the cylinder, thus causing nearly all the work to be done in one end.

Correction of Fault. By following the directions as previously given for valve setting by measurement or by indicator, it becomes a comparatively small matter to correct the trouble.

CORLISS VALVE GEAR

Description. The Corliss valve gear is the most widely known of all the types of so-called "drop cut-off" valve gears. It is more economical than most other types from the standpoint of steam consumption but, on account of its peculiar construction and multiplicity of parts, is not adapted for high-speed work, say, above 100 revolutions per minute. Directions for setting a Corliss valve gear have been presented earlier in this text and need not be repeated here, but there is a word of caution which should be emphasized.

Possible Troubles. The rods connecting the steam valve arms with the dash pots should be adjusted so that when down

as far as they will go and with the wrist plate in its extremes of travel the stud die on the valve arm will just clear the shoulder on the hook die. If the rod is left too long, the steam valve stem will probably be bent, the valve arm broken, or the dash pot rod bent or broken. It may happen that the jar from the action of the dash pot will cause the dash pot rod to become loosened while in service. If this occurs the parts just mentioned may be broken in a manner similar to that when the rod is left too long in setting. Again, if the dash pot rod is left too short, the hook will not engage and, consequently, the valve will not open.

STEPHENSON VALVE GEAR

Extent of Use. The Stephenson gear, or *link motion*, as it is commonly called, is one of the oldest and best known types of reversing gears in use in the United States. For a great many years it was used almost to the exclusion of all other types of gears on American locomotives. Of recent years, however, its use has declined until today we find only comparatively few American locomotives equipped with the Stephenson reversing gear. The use of this gear is not confined entirely to locomotive service. In fact, it is made use of on steam engines in many classes of service, such as, steam tractors, steam road rollers, stationary engines, and hoisting engines.

Characteristics. *Increase of Lead in Open-Rod Construction.* One of the characteristics of the Stephenson reversing gear is that the lead of the valve increases from full to mid gear for open-rod construction and decreases from full to mid gear for crossed-rod construction. The crossed-rod construction is seldom used on engines unless service conditions are such as to make necessary its manipulation by the use of the reversing lever. The feature of increasing lead from full to mid gear, under certain conditions, is desirable on locomotives used for passenger service. In such instances the engineer will usually start the train with the reverse lever at or near the full gear position where the lead is a minimum and as the speed increases will bring the reverse lever nearer and nearer the central position where the lead is greater. This feature considered by itself is desirable since for best working conditions the lead should increase with the speed.

Back-Up Eccentric. One very desirable feature of the Stephenson gear is that it may be set to secure almost any steam distribution desirable. This is accomplished by making use of the "back-up" eccentric. Applying this method to setting the valves will, of course, disarrange the reverse, or "back-up", conditions but the "go-ahead" conditions can be almost perfectly secured.

Possible Troubles. *Lost Motion in Driving Boxes.* In the use of Stephenson gears on locomotives there is one condition which frequently arises but is rarely considered. The condition referred to is the development of lost motion in the driving boxes. In such cases, the eccentric being attached to the axle, the full amount of this lost motion is delivered to the valve with the link working in full gear. In certain other types of gears this condition would produce but very little change in the movement of the valves.

Effect of Vertical Motion of Engine. Another condition which affects the steam distribution when a Stephenson gear is used is the vertical motion of the engine on its springs caused by irregularities in the track.

Setting Valve. In setting the valve on an engine using the Stephenson gear, the fundamental principles involved are exactly the same as those given for the setting of a plain slide valve gear. We need to keep constantly in mind, however, that there are two eccentrics and two eccentric rods to deal with instead of one.

Typical Plain Slide Valve Setting. As an example let us consider the case of an engine fitted with a plain slide valve gear. Suppose it is desired to give the valve a lead of $\frac{1}{32}$ inch on both the head and crank ends. An examination of the valve discloses the fact that the lead on the head end is $\frac{1}{8}$ inch and that on the crank end is $\frac{1}{32}$ inch, which is the desired amount. The problem is to reduce the lead on the head end $\frac{3}{32}$ inch without disturbing the lead on the crank end. This problem can be solved by reducing the lead on the head end $\frac{3}{64}$ inch by changing the length of the valve rod and an additional $\frac{3}{64}$ inch by changing the angle of advance of the eccentric on the shaft. If the work is carefully done the results should show a lead of $\frac{1}{32}$ inch on both ends, unless the angularity of the eccentric rod is a very considerable amount.

Stephenson Valve Setting. Now suppose that the simple slide valve gear on this engine has been replaced by a Stephenson revers-

ing gear and that an examination of the valve with the reverse lever in full gear position shows the lead on the head end to be $\frac{1}{8}$ inch and that on the crank end $\frac{1}{32}$ inch for the forward position of the reverse lever, while for the backward position the leads on both the head and crank ends are found to be correct, namely, $\frac{1}{32}$ inch. In this case, the same as before, it is desired to secure a lead of $\frac{1}{32}$ inch on each end when the reverse lever is in both the forward and backward positions. To accomplish this with the reverse lever in the full forward position, it will be necessary to reduce the lead $\frac{3}{64}$ inch by changing the length of the eccentric rod and an additional $\frac{3}{64}$ inch by changing the position of the eccentric on the shaft. If the work is carefully done the desired results will be approximately secured.

Differences in the Two Settings. In the example just presented it should be noted that in the case of the simple gear the adjustments were made on the valve rod and eccentric, while in the case of the Stephenson gear they were made on the eccentric rod and eccentric. This correction of one-half the error on the eccentric and one-half on the eccentric rod, instead of on the valve rod, is necessary in order to permit the conditions on the reverse direction to remain unchanged. Other adjustments of a like nature can be made in a similar manner.

Variation in Conditions. Unfortunately, in practice, the Stephenson reversing gears are not always constructed so as to permit all the adjustments mentioned above. In such instances a compromise will have to be made.

WALSCHAERT GEAR

Extent of Use. The Walschaert gear has been used abroad for many years but never attained prominence in this country until ten or twelve years ago. It represents the most satisfactory type of radial reversing gear now in service in the United States. It is now being equipped on approximately 80 per cent of all new American locomotives, the remaining 20 per cent being fitted with the Stephenson gear. Its use, however, is confined almost exclusively to locomotive service.

Comparison with Stephenson Gear. One of the chief advantages of the Walschaert gear is the accessibility of all the parts and

the comparative ease with which repairs can be made. The parts of the gear being located outside, the space below the boiler may be used for other parts not so necessarily accessible. The chief point in which the Walschaert gear differs from the Stephenson gear on the action of the valve is that the former gives a constant lead for all positions of the reverse lever. Both gears are adaptable for use with any form of locomotive valve yet designed. The usual construction of the Walschaert gear is such as to permit little or no adjustments being made on the road. It is unusually free from any inclination of the parts to cause trouble through heating. Cases are known where improperly designed gears gave some trouble by the eccentric rod pins heating due to the twisting effect between the driving wheels and engine frame caused by unusual conditions of track and service. This, however, is a matter easily corrected.

Lost motion in the driving boxes produces much less effect on the motion of the valve when a Walschaert gear is used than when a Stephenson gear is employed. Neither does the up-and-down motion of the engine on its springs affect the steam distribution unless the connection of the eccentric rod to the link foot is placed at too high a point above the center line of the axle. In all well-designed Walschaert gears it is necessary that the trunnion upon which the link oscillates be fixed at an unvarying distance from the cylinder. In the fulfillment of this requirement it will be observed that the link bracket is invariably attached to the guide bearer, or yoke, and the slide for the valve stem is mounted on the upper guide bar. In some types of locomotives the construction is such that a large cast-steel bracket is laid across, joining the bars of the engine frame on both sides just back of the guide yoke, which acts as a frame binder and brace and a carrier for the link bracket. In still another type, the large casting is bolted to the guide yoke as well as the frame, thus forming a most substantial construction.

Repairs. With the Walschaert gear in service, if a break occurs within the valve gear, the difference in time consumed in making the temporary repairs necessary to get the engine moving under its own steam is greatly in its favor. This is one of the principal reasons for its adoption, since it means less time lost in delays.

REVIEW QUESTIONS

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM ENGINES

PART I

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1. When and by whom was the first commercially successful steam engine built? What was the most serious objection to this engine?

2. In what way did Newcomen improve the steam engine?

3. What important improvements did James Watt make in the steam engine?

4. Describe in your own words the first pumping engine made famous because of its high economy.

5. State the function of the engine frame.

6. Show by sketch the construction of a plain slide-valve steam-engine cylinder, and explain its operation.

7. What are some of the things to be observed in the construction of the piston, piston rod, and crosshead of a steam engine?

8. Name the different forms of packings used on the piston rod of a steam engine and state which is the most preferable and why.

9. State the chief advantage of a piston valve over a plain slide valve.

10. Name the two general classes of connecting rods now in use. Of what material are they constructed?

11. Why are counterweights placed on the crank-disks of high speed engines?

12. What two functions does the flywheel of an engine usually perform?

13. State the distinction between simple and compound engines. What advantages are obtained by their use?

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM ENGINES

PART II

1. Name two types of condensers now in use and state the advantages secured by their use.

2. The steam from an engine delivering 100 horsepower is condensed by means of a jet condenser. If the engine uses 24 pounds of steam per horsepower per hour, how much condensing water will be required per hour (theoretical), if the water enters at 55°F. and leaves at 118°F., the vacuum gauge reading 13.7 pounds absolute?

3. In what way is it possible for a condenser to increase the power of an engine?

4. Generally speaking, what factors determine whether it is economical to operate an engine condensing or noncondensing?

5. When are cooling towers used? Describe the general construction and operation of a large cooling tower.

6. What danger arises from using the condensed steam from a surface condenser as feed water for the boiler?

7. At what position of the crank is the turning moment on the main shaft a maximum?

8. Explain by means of a diagram that the crank effort is more nearly constant with two or more cranks on the main shaft than with one.

9. Explain why the turning moment varies on the shaft of a single cylinder engine.

10. What is the function of the flywheel when considering speed regulation?

11. Which class of engine needs the larger flywheel and why?

REVIEW QUESTIONS

ON THE SUBJECT OF

STEAM ENGINE INDICATORS

1. Enumerate the essential parts of a steam engine indicator.

2. Of what value is the steam engine indicator in the design and operation of the steam engine?

3. Can the steam engine indicator be used in indicating gas engines? What changes, if any, are necessary in the indicator?

4. State the essential differences between the Crosby and the Thompson indicators.

5. In the study of an indicator diagram, state what is meant by the atmosphere line, clearance line, absolute pressure line, admission line, steam line, expansion line, release line, back pressure line, and compression line.

6. Name the essentials of a good indicator.

7. What is meant in speaking of a 60-pound indicator spring?

8. State what is meant by an engine having a mean effective pressure of 48.5 pounds.

9. How can too late admission be detected from the indicator card and how may it be remedied?

10. Why should indicator springs be calibrated occasionally and how may it be accomplished?

11. In drilling and tapping an engine cylinder for indicator connections, what precautions should be observed?

12. State the effect of long pipe connections on the indicator card and why they should be avoided.

13. What is meant by the piston displacement of an engine cylinder and why is the crank end smaller than that of the head end of the cylinder?

14. Sketch an indicator card and locate and name the four events of the stroke.

REVIEW QUESTIONS

ON THE SUBJECT OF

VALVE GEARS

1. Name the various parts which go to make up the valve gear of a steam engine.

2. What is the function of the valve gear?

3. In the case of a plain slide valve, why is the valve seat usually constructed so that the valve travels over the edge of the seat a small distance?

4. What is meant by the *throw of an eccentric?* What will be the throw of an eccentric if the valve travel is $3\frac{1}{2}$ inches?

5. When is a plain slide valve said to be in mid-position?

6. Sketch a section through a plain slide valve showing the valve seat and ports and indicate the amount of inside and outside lap.

7. What is meant by inside and outside lap, and how may they be determined in a given case?

8. Given the case of a plain slide valve having lap, would the eccentric be placed more or less than ninety degrees with the crank arm? State your reasons.

9. Under what conditions would the eccentric be placed at right angles to the crank arm? Would this arrangement give economical results? If not, why?

10. What is *lead*, and why is it desired?

11. What effect does increasing the outside lap have on the steam distribution?

12. What effect does increasing the inside lap have on the steam distribution?

13. How do changes in the angular advance and eccentricity affect the steam distribution?

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